3.1. Trufin Tubes in Condensing Heat Transfer

3.1.1. Modes of Condensation

Condensing is the heat transfer process by which a saturated vapor is changed into a liquid by means of removing the latent heat of condensation.

Four basic mechanisms of condensation are generally recognized: dropwise, filmwise, direct contact, and homogeneous. In dropwise condensation, the drops of liquid form from the vapor at particular nucleation sites on a solid surface, and the drops remain separate during growth until carried away by gravity or vapor shear. In filmwise condensation, the drops initially formed quickly coalesce to produce a continuous liquid film on the surface through which heat must be transferred to condense more liquid. In direct contact condensation, the vapor condenses directly on the (liquid) coolant surface which is sprayed into the vapor space. In homogeneous condensation, the liquid phase forms directly from super saturated vapor, away from any macroscopic surface; it is however generally assumed that, in practice, there are sufficient numbers of dirt or mist particles present in the vapor to serve as nucleation sites.

While dropwise condensation is alluring because of the high coefficients reported, it is not considered at this time to be suitable for deliberate employment in process equipment. Generally, contaminants must be continuously injected into the vapor, or special materials (often of low thermal conductivity) employed. Even so, the process is unstable and unpredictable, and of questionable efficacy under conditions of high vapor velocity and industrial practice.

Direct contact condensation is a very efficient process, but it results in mixing the condensate and coolant. Therefore, it is useful only in those cases where the condensate is easily separated, or where there is no desire to reuse the condensate, or where the coolant and condensate are the same substance. Homogeneous condensation is primarily of concern in fog formation in equipment and is not a design mode.

Therefore, all subsequent references to condensation will mean filmwise condensation, in which the heat transfer surface is covered with a thin film of condensate flowing under the influence of gravity, vapor shear, and/or surface tension forces.

The necessary equations for calculating the heat transfer and pressure drop for condensing will be developed later in this Chapter. The case for in-tube condensing will be studied first then extended to cover condensing outside Trufin tubes.

3.1.2. Areas of Application

In Chapter 1, it was pointed out that it is usually advantageous to use Trufin when one of the film heat transfer coefficients is significantly smaller than the other. The lower coefficient tends to control the magnitude of $U$, the overall heat transfer coefficient, and therefore the size of the heat exchanger. Hence, if Trufin is used, with the low coefficient fluid in contact with the higher heat transfer area of the fin, the total amount of tubing is reduced compared to the plain tube case; therefore the overall size of the heat exchanger is also reduced.

The best design is generally obtained if the thermal resistances of the two fluid heat transfer processes are approximately equal. This condition is obtained when:
In a large number of condensing applications in the process and refrigeration industries, especially where water cooling is used, the value of $h_i/A_i$ ranges from 2 to 5 or even 10. Since low- and medium-finned Trufin have $A_o/A_i$ values from about 3 to 7, these tubes are often found to afford substantial savings in overall heat exchanger size and cost. In these applications the condensation takes place on the outside (fins) of the tubes.

In other applications where air is used as the cooling medium, the air side heat transfer coefficients are much lower than the condensing coefficients. High-fin Trufin is used in these cases with the condensing taking place inside the tubes and the high outside area placed on the air side.

### 3.1.3. Types of Tubes Available

1. **Type S/T Trufin® Low-Finned Tube**
   Tubes of this type are made with 16 to 40 fins per inch and fin heights of approximately 1/16 inch. The diameter over the fins is equal to or less than the plain end diameter to allow the tube to be inserted through a tubesheet.

2. **Medium-Finned Trufin**
   These tubes are characterized by having 11 fins per inch and fin heights of 1/8 inch. The tubes can be supplied with plain ends of a smaller diameter than the finned section (type W/H) or with belled ends suitable for rolling into tubesheets (type S/T).

3. **Type S/T Turbo-Chil® Finned Tubes**
   The outer surface of these tubes is similar to standard type S/T Trufin. In addition, the inner surface of the tube is provided with integral spiral ridges which enhance the internal heat transfer coefficient.

4. **Koro-dense®**
   This tube is a corrugated rather than a finned tube but is mentioned here because of its advantageous application in steam condensing. Two types are available: MHT, a medium corrugaion severity affording maximum tube side performance if pressure drop permits, and LPD, a low corrugaion severity for use when tube-side pressure drop is limiting.

   Condensation generally takes place on the outside surface of the above tubes.

5. **High-finned Trufin**
   Tubes of this type are made in both copper and aluminum. Fin counts range from 5 to 11 fins per inch with fin heights as high as 5/8 inch. The aluminum finned tube can be supplied with liners of various other metals.
3.2. Condensation of Vapor Inside High-Finned Trufin Tubes

3.2.1. Vapor-Liquid Two-Phase Flow

1. Phase Relationships of Two-Phase Condensing Flows. Condensation of a vapor is frequently carried out inside the tubes of an air-cooled heat exchanger employing high-finned Trufin tubes. Usually the tubes are horizontal or nearly so, but occasionally inclined or vertical orientations are used. The flow of a vapor-liquid two-phase mixture is a good deal more complicated than single phase flow, and the correlations for heat transfer and pressure drop correspondingly less accurate. Since in general the two-phase heat transfer coefficients are quite high, this inaccuracy is not a serious matter for heat transfer in air-cooled condensers. However, the pressure drops are also quite high and care is required in interpreting the predicted values into design considerations.

The first way to characterize a condensing two-phase flow is by its composition. Five cases are discernible:

a. The liquid and the gas are different pure components. The usual example is an air-water mixture, which is not a common industrial problem, but is very important because a great deal of what is known about two-phase flow has been determined on this system. While in general this information can be carried over to condensation and some boiling work, there are important differences that must be recognized and allowed for. Thermodynamically, the pressure and temperature can be independently varied over wide ranges in this system.

b. The liquid and gas (vapor) are the same pure component. This is a common case in condenser design, occurring for example in condensers on columns separating and/or purifying a product. The pressure-temperature relationship in this case is the vapor pressure curve for the component.

c. The liquid and gas (vapor) are multi-component mixtures, each containing some of each of the components. The thermodynamic relationships are more complex, the temperature, for example, being variable over a range of values at a given pressure, but with a changing ratio of total liquid to total vapor and with changing composition of each phase. Prediction of the amount and composition of each phase is relatively well understood and easily done in a few cases, such as mixtures of light hydrocarbons; other cases require laboratory thermodynamic data.

d. Like case (b) or (c), but with a non-condensable gas present. The most common example cited here is air in steam, but there are also applications as condensation of a solvent from an inert stripping gas.

e. Intermediate between (c) and (d), typified by condensation of light gasoline fractions from a wet natural gas. Depending on the composition of the gas and the pressure and temperature of the condensation, some of the methane and ethane present will dissolve in the liquid, and a small amount of the heavier components will be present in the vapor.

2. Characterization of Two-Phase Vapor-Liquid Flows. This section will concentrate on case (b), condensation of a pure component, but many of the equations and procedures developed will be applicable to the other situations, as discussed later. There are a number of quantitative parameters of two-phase flows that characterize the flow and must be defined:
a. Weight flow rates: The weight flow rates of each phase (in typical units of lb/hr), \( w_v \) and \( w_l \), are generally either known or computable from process specifications. In a condensing flow, the total weight flow rate remains constant, the liquid rate increasing as the vapor rate decreases. The flow mechanisms of the condensation process may change as the ratio changes from entrance to exit. It is convenient to non-dimensionalize the ratio of the two phases by defining the quality \( x \) as:

\[
x = \frac{w_v}{w_v + w_l}
\]  
(3.1)

b. Volume flow rates: The volume flow rates are occasionally useful quantities and are defined as:

\[
\vartheta_g = \frac{w_v}{\rho_v}, \quad \vartheta_g = \frac{w_l}{\rho_l}
\]  
(3.2)

c. Mass velocities: The mass velocity of the total two phase stream, \( G \) (lb/hr ft\(^2\)) is constant in a constant cross-section conduit. It is defined as:

\[
G = \frac{w_v + w_l}{S}
\]  
(3.3)

where \( S \) is the cross-sectional area of the conduit.

d. Superficial mass velocities: The superficial mass velocity of a given phase is defined as:

\[
G_v = \frac{w_v}{S} = G_x
\]  
(3.4)

\[
G_l = \frac{w_l}{S} = G(1-x)
\]  
(3.5)

The entire cross-sectional area of the conduit is used in calculating \( G_v \) and \( G_l \) even though each phase alone occupies only a portion of the cross section. These quantities have no fundamental significance, but they are convenient and useful parameters for correlating two-phase flow data.

e. Velocities: The true mean velocities of each phase are not simply or fundamentally related to the other characterizing parameters of the system. The liquid, being generally preferentially located near the walls of the conduit and being more viscous and dense than the vapor, in general flows more slowly, giving rise to the phenomenon of “slip.” If \( V_v \) is the actual mean velocity of the vapor in (ft/hr) at a given point in the conduit and \( V_l \), is the corresponding value for the liquid, \( V_v \) is usually greater than \( V_l \). It is customary to define a quantity, the slip ratio, as \( V_v / V_l \); as noted above, the slip ratio is almost always greater than 1 and can reach 10 or 20 or even more. If \( V_v = V_l \), the flow is said to be homogeneous; this is not a very realistic case, but it is simple to use in that it permits densities, velocities, and weight and volume fractions to be directly related to each
other. In the lack of better information, homogeneous flow may be assumed taking precaution against non-conservative consequences of this assumption.

f. Phase volume fractions: The actual volume occupied by the vapor phase divided by the total conduit volume is called the vapor phase volume fraction, \( R_v \), or perhaps more commonly, the void fraction. The liquid volume fraction, \( R_l \), is equal to \( 1 - R_v \). For a steady flow, or one averaged over a sufficient period of time, these are also the fractions of the cross-sectional area that are occupied by the respective phases. By use of continuity, we may write

\[
R_v = \frac{S_v}{S} = \frac{\frac{w_v}{\rho_v V_v}}{x + \frac{\rho_v V_v}{\rho_l V_l}} = \frac{x}{x + \rho_v V_v / \rho_l V_l (1 - x)}
\]  

In these terms the slip ratio is given by

\[
s = \frac{V_v}{V_l} = \frac{\rho_l}{\rho_v} \frac{1 - R_v}{R_v} \frac{x}{1 - x}
\]  

3. Correlations of the Vapor Volume Fraction. It is evident that data on the vapor volume fractions are needed in order to calculate some of the characteristic quantities of a two-phase flow. Many experimental studies have been made to determine values of \( R \), and a number of correlations have been proposed (Ref. (1) surveys the work done up to about 1965). A generally usable correlation which seems to give reasonable results over the range of process applications was proposed by Martinelli and co-workers, mostly based on air-water systems at about atmospheric pressure. The correlation for volume fractions proposed by Martinelli and Nelson (2) seems to work as well as any other correlation proposed over the entire range of data available. The original correlation was intended for steam-water systems, but is easily generalized in terms of the reduced pressure. The modified correlation is given here as Fig. 3.1. The abscissa of Fig. 3.1 is

\[
x_{R} = \left( \frac{1 - x}{x} \right)^{0.57} \left( \frac{\rho_v}{\rho_l} \right)^{0.11}
\]  

The ordinates are \( R \), and \( Re \); the parameter is the reduced pressure of the vapor

\[
P_{R} = \frac{p}{p_{crit}}
\]  

The form of the parameter \( x_{R} \) is deduced by Martinelli from a particular model of two phase flow but it seems to be remarkably generally applicable. The limiting curve for \( P_{R} = 0.005 \) is taken from the
experimental results at atmospheric pressure for air-water and the curve for $P_R = 1$ is obtained from the requirement that the slip ratio go to unity as the two phases become more similar. The curves for other reduced pressures are interpolated by eye.

4. Regimes of Two-Phase Flow. To this point, we have said nothing of the actual physical appearance of a two-phase flow. In some respects, such as void fraction calculations for process condensers, the physical description of the flow is relatively unimportant. In other regards, it is of some importance. The types and the detailed description of two-phase flow configurations (or regimes) depends upon the relative and absolute quantities and the physical properties of the fluids flowing, the geometric configuration of the conduit, and the kind of heat transfer process involved. The flow regimes observed by
Alves (3) in his study of air-water flows in horizontal tubes are diagrammed in rather idealized form in Fig. 3.2. The chief difference between flow regimes studied in a non-heat transfer situation and those existing during condensation is that a liquid film exists on the entire surface of the conduit during condensation; however, we may presume that this film of draining condensate does not cause any vital difference in the interaction between the vapor and the main inventory of liquid.

We may view the flow regime as a consequence of the interaction of two forces-gravity and vapor shear acting in different directions. At low vapor flow rates, gravity dominates and one obtains stratified, slug-plug, or bubble flow depending upon the relative amount of liquid present. At high vapor velocities, vapor shear dominates, giving rise to wavy, annular, or annular-mist flows. It would be desirable to have some way to predict the flow regime a priori, and many attempts have been made to do this in a general and consistent way. No attempt has succeeded, but the work of Baker (4) is considered to be generally the best available even though it is a dimensional representation and defies explanation in fundamental terms. The Baker map is shown in modified form as Fig. 3.3 and is useful in giving a general appreciation of the general kind of flow regime existing under given conditions.

During the course of condensation, the relative ratios of vapor and liquid change; these changes may be followed conveniently on the Baker map with the aid of Fig. 3.4, which is used in the following way: Calculate the values of the abscissa and ordinate on the Baker map corresponding to the half-condensed point \( G_v = \frac{l}{G} \). Plot this point on Fig. 3.3. Place the 0.5 point (circled) of the curve of Fig. 3.4 on top of the point plotted on Fig. 3.3 (it is helpful if a transparency of Fig. 3.4 is used) so that the coordinates are parallel on the two figures. Then the curve on Fig. 3.4 traces the sequence of flow regimes as a function of vapor quality from pure vapor at the left infinity point to pure liquid at the bottom in infinity point.

It may be assumed that condensing flow patterns in tubes inclined slightly downwards (in the direction of flow) are similar to those in horizontal tubes. Condensation is seriously reduced in tubes inclined slightly upwards (5) and care must be taken to insure that this does not occur, often by deliberately designing the condenser to a slight (2-3°) downward inclination.

For two-phase flow inside vertical tubes, the stratified and wavy flow regimes cannot exist, and the flow regimes generally recognized in this case are bubble (low vapor rate), slug, annular, and annular with mist. There is no generally recognized flow regime map for condensing in vertical tubes at the low heat fluxes generally characteristic of air-cooled condensers. This is not a serious deficiency in predicting heat transfer coefficients and the same is also true of condensing in steeply inclined tubes.
5. Pressure Drop in Two-Phase In-Tube Flow.
Pressure changes in two-phase flow arise from three sources.

a. Friction loss, which always causes a pressure decrease in the direction of flow. The pressure loss due to friction in a two-phase flow is generally much higher than in a comparable single phase flow because of the roughness of the vapor-liquid interface. The pressure gradient due to friction depends upon local conditions, which change in a condensing flow. Therefore, the total pressure effect from friction depends upon the path of condensation.

b. Momentum effects, resulting from a change in the velocity, and hence kinetic energy, of the stream. The pressure change from this cause is negative (pressure loss) if the flow accelerates as in boiling and positive (pressure gain) if the flow decelerates, as is the usual case in condensation. The total effect on pressure from this cause depends only upon terminal conditions, though in some cases it may be necessary to calculate the local pressure gradient contributed by velocity changes.
c. Hydrostatic effects, resulting from changes in elevation of the fluid. The pressure from this source alone always decreases in an upward direction and is zero in a horizontal tube. The local gradient depends upon the local density; therefore, it is generally necessary to calculate this contribution along the path of condensation.

The algebraic sum of the three contributions is equal to the net total pressure effect. In process applications, the friction loss usually predominates in horizontal configurations and is usually comparable to the hydrostatic effect in vertical designs. However, in very high velocity condensing flows, the momentum effect may be greater than the other contributions, resulting in a pressure rise from inlet to outlet (e.g., Ref. (6)).

Calculation of the pressure loss due to friction may be made using the Martinelli-Nelson correlation (2). First, we calculate the pressure gradient as if only the liquid were flowing in the conduit.

\[
\left( \frac{dp}{d\ell} \right)_{f, l} = -\frac{2f_l G_l^2}{g_c \rho_l d_l}
\]  

(3.11)

where \( f_l \) is read from the Fanning friction factor chart or Eq.(3.13) at a Reynolds number calculated from:

\[
Re_l = \frac{d_l G_l}{\mu_l}
\]  

(3.12)

if \( Re_l > 2100 \). If \( Re_l < 2100 \), use the vapor-phase based correlation given further on in this section. The smooth tube correlation may be used though it is more conservative to use the correlation for the relative roughness of the actual surface at high Reynolds numbers. Over the usual range of process applications, \( f_l \) may be computed from

\[
f_l = \frac{0.078}{\left( \frac{d_l G_l}{\mu_l} \right)^{1/4}}
\]  

(3.13)

The pressure gradient due to friction for the two-phase flow is then calculated from

\[
\left( \frac{dp}{d\ell} \right)_{f, TPF} = \Phi_{l/l} 2 \left( \frac{dp}{d\ell} \right)_{f, l}
\]  

(3.14)

where \( \Phi_{l/l} \) is read from Fig. 3.5 at the appropriate value of

\[
x_l = \left( \frac{1-x}{x} \right) \left( \frac{\rho_v}{\rho_l} \right)^{0.57} \left( \frac{\mu_l}{\mu_v} \right)^{0.11}
\]  

(3.15)

If \( Re_l < 2100 \), and \( Re_v > 2100 \), the calculation proceeds similarly, but based on the vapor phase.
\[
\left( \frac{dp}{d\ell} \right)_{f,v} = -\frac{2f_v G_v^2}{g_c \vartheta \varphi d_i}
\]  

(3.16)

where \( f_v \) is taken from the Fanning chart at

\[ Re_v = \frac{d_i G_v}{\mu_v} \]  

(3.17)

or from

\[ f_v = \frac{0.078}{\left( \frac{d_i G_v}{\mu_v} \right)^{1/4}} \]  

(3.18)

\( \Phi_{tt}^{-2} \) is again read from Fig. 3.5 at the appropriate value of \( x_m \) and the two-phase friction pressure drop is given by

\[
\left( \frac{dp}{d\ell} \right)_{f,TPF} = \Phi_{tt}^{-2} x_m^{1.75} \left( \frac{dp}{d\ell} \right)_{f,v}
\]  

(3.19)

If both \( Re_v \) and \( Re_v \) are below 2100, an approximate value may be obtained by using the laminar flow friction factor equation for either phase:

\[ f_l \text{ or } v = \frac{16}{Re_l \text{ or } v} \]  

(3.20)

However, in this case, the pressure drop will probably be so low as to be unimportant.

The pressure effect due to momentum changes is given by

\[
\left( \frac{dp}{d\ell} \right)_{m,TPF} = -\frac{G}{g_c} \frac{d \left[ v_t (1-x) + v_v x \right]}{d\ell}
\]  

(3.21)

However, the total effect can be calculated from inlet and outlet conditions only:

\[
\Delta p_{m,TPF} = \frac{G}{g_c} \left[ v_t (1-x) + v_v x \right]_{inlet} - \left[ v_t (1-x) + v_v x \right]_{outlet}
\]  

(3.22)

If a dry saturated vapor enters and is completely condensed, Eq. (3.22) reduces to

\[
\Delta p_{m,TPF} = \frac{G}{g_c} \left[ v_v \text{inlet} - v_t, \text{outlet} \right] = \frac{G^2}{g_c} \left[ \frac{1}{\rho_v} - \frac{1}{\rho_t} \right]
\]  

(3.23)
Fig. 3.5 Correlation of $\Phi_{in}$ as a Function of $\sqrt{V/\Delta t}$.

- $P_R = 1.0$
- $P_R = 0.6$
- $P_R = 0.3$
- $P_R = 0.005$
Recall that for condensation this is a pressure increase.

The hydrostatic pressure effect is calculated from

\[
\frac{dp}{d\ell}_{g,TPF} = \rho_{eff} g \cos \Theta \frac{d\ell}{g_c} 
\]

where \( \rho_{eff} \) is given by Eq. (3.8) and \( \Theta \) is the angle between the vertical and the axis of the tube (\( \Theta = 0 \) for a vertical tube; \( \Theta = \pi/2 \) for a horizontal tube, see Fig. 3.6). The hydrostatic contribution to the pressure is always negative in the upward direction.

The total local pressure gradient is the algebraic sum of the three effects:

\[
\left( \frac{dp}{d\ell} \right)_{T,TPF} = \left( \frac{dp}{d\ell} \right)_{f,TPF} + \left( \frac{dp}{d\ell} \right)_{m,TPF} + \left( \frac{dp}{d\ell} \right)_{g,TPF} 
\]

The total pressure difference from one end of the condenser to the other must be found by integration over the quality range

\[
\Delta p_{TPF} = \int_{x_i}^{x_f} \left[ \frac{dp}{d\ell} \right]_{T,TPF} \left[ 1 + \frac{dx}{d\ell} \right] d\ell
\]

Exact numerical evaluation of this integral is a rather tedious trial-and-error process, even on a computer, because the properties and the local quality depend upon the local pressure implicitly. For a condenser, however, one is usually concerned with complete condensation (\( x_1 = 1; x_0 = 0 \)) and with a very nearly linear condensation rate, \( (dx/d\ell) = 1/L \). Then, a good approximation to \( \Delta p_{TPF} \) is obtained by a "pseudo-Simpson" rule:

\[
\Delta p_{TPF} = \Delta L \left[ \frac{25}{96} \left( \frac{dp}{d\ell} \right)_{TPF, x=0.1} + \frac{23}{48} \left( \frac{dp}{d\ell} \right)_{TPF, x=0.5} \right]
\]

6. Flooding Effects in Upward Flow. In some applications, variously termed reflux or knockback condensers, the vapor flows upward in the tube while the liquid flows downward on the walls. In this case, the vapor shear subtracts from the gravitational force causing the liquid to flow downwards, leading to a thickening of the film; if the film is in laminar flow, the heat transfer coefficient is also reduced, an effect analyzed by Nusselt (7). In general, the effect is rather to cause early turbulence, which does thicken the film, but also increases the coefficient. If the film would be turbulent even in the absence of vapor shear, the effect of vapor shear on heat transfer is not clear and probably not of critical importance.

The main concern in knockback condensers is that flooding be avoided, i.e., that at no point in the condenser should the vapor shear be equal to the gravitational force of the liquid. The critical point is at the entrance to the lower end of the tube when both the liquid and vapor flow rates are maximum. Soliman, Schuster, and Berenson (8) have analyzed the problem in very fundamental terms, but the best design correlation for flooding is by Diehl and Koppany (9). In summary, their correlation for incipient flooding is
\[ V'_{v} = F_1 F_2 \left( \frac{\sigma}{\rho_v} \right)^{0.5} \]
\[ V'_{v} = 0.71 \left[ F_1 F_2 \left( \frac{\sigma}{\rho_v} \right)^{0.5} \right]^{-1.15} \]

where

\[ V'_{v} = \text{superficial flooding velocity of vapor, ft/sec} \]

\[ F_1 = \left( \frac{12d_t}{(\sigma/80)} \right)^{0.4} \]
\[ F_2 = \left( \frac{G_v}{G_i} \right)^{0.25} \]

The equation is dimensional, so that it is essential to use the units specified (\( \rho_v \) in lb/ft\(^3\), \( d \) in ft, \( \sigma \) in dyne/cm). Since this is for incipient flooding, conservative design requires that the maximum design vapor velocity be somewhat below this; if the conditions are within the fairly wide range represented by the data in the Diehl-Koppany correlation, design to 70 percent of the predicted flooding velocity appears safe. Otherwise, the velocity should be limited to 50 percent of that calculated.

In a few cases, condensers have been designed so that the vapor shear is great enough to carry the condensate up and out of the condenser. Total condensation is impossible in this case since there must be sufficient vapor flow at the top of the condenser to carry the liquid out. It has been observed that, as one increases the vapor flow above the Diehl-Koppany limit for flooding, the amount of liquid carryover increases slowly until virtually complete carryover occurs at about three times the flooding velocity. This value then becomes the design case at the top of the condenser for relatively foolproof operation. For intermediate values, the liquid that drains back must be carried up into the condenser again and blown through, a situation that offers opportunity for slugging, maldistribution, and generally unstable operation.

3.2.2. Condensation Heat Transfer

Laminar Film Condensation of a Pure Component in a Vertical Tube. Laminar film condensation of a pure component from a saturated vapor was among the first heat transfer problems to be successfully analyzed from a fundamental point of view. The definitive work is by Nusselt (7) in two papers published two weeks apart in 1916. The analysis is readily available in a number of books, of which Jakob (10) and Kern (11) may be especially recommended. The original Nusselt analysis applies specifically to laminar flow of a condensing film on a vertical surface. However, it is possible to generalize the approach to apply to a number of other cases, including in-tube condensation in horizontal tubes. For this reason, it is worthwhile to examine the analysis in some detail here. There are a number of assumptions implicit in the basic Nusselt model. The key assumption is that the liquid film (Fig. 3.7) is in laminar flow and its hydrodynamics are controlled by the viscous terms in the Navier-Stokes equations. This allows the neglect of the inertial or kinetic energy terms and yields a simple equation relating film thickness and velocity profile to gravity and liquid viscosity. Relaxing this assumption yields complex equations that must be solved numerically; this has been done by several authors and the results indicate that the more
complex solution is negligibly different from the simple one for most process applications. Other assumptions of the Nusselt model include:

1. Saturated vapor.

2. The liquid and the vapor have the same temperature \( T_{\text{sat}} \) at the interface. (No interfacial resistance.)

3. Heat is transferred by conduction only through the liquid film.

4. The temperature profile is linear through the liquid film.

5. The liquid and the solid surface are at the same temperature at their interface.

6. The solid surface is isothermal.

7. The liquid properties are not a function of temperature.

8. The vapor exerts neither shear nor normal stresses on the liquid surface.

9. The liquid has zero velocity at the liquid-solid interface (no-slip condition).

10. The sensible heat of subcooling the liquid is negligible compared to the latent heat load.

Before discussing the validity of these assumptions, let us look at the resulting equations. The local value of the film heat transfer coefficient at a distance \( x \) from the start of condensation is

\[
h_x = \left[ \frac{k_{\ell} \rho_{\ell} (\rho_{\ell} - \rho_v) \lambda g}{4 \mu_{\ell} (T_{\text{sat}} - T_w) x} \right]^{1/4}
\]

A far more useful quantity is the average coefficient for a surface of length \( L \), which we identify for convenience as the condensing coefficient \( h_c \):

\[
h_c = \int_0^L h_x \, dx = 0.943 \left[ \frac{k_{\ell} \rho_{\ell} (\rho_{\ell} - \rho_v) \lambda g}{\mu_{\ell} L (T_{\text{sat}} - T_w)} \right]^{1/4}
\]

Fig. 3.7 Nusselt Condensation on a Plane Vertical Surface.
Note that the heat transfer coefficient predicted by (3.33) decreases as \( L \) and \( (T_{\text{sat}} - T_w) \) increase. This is due to the increased resistance to conduction offered by a thickened film.

The derivation of Eq. (3.33) was carried out in terms of a vertical plane surface. Since the condensate film is so thin compared to typical tube diameters, the result is applicable to condensation on the inside or outside of vertical tubes if the other assumptions of the derivation are satisfied. Strictly speaking, the dependence of \( h_c \) on \( L \) and \( (T_{\text{sat}} - T_w) \) violates two of the assumptions underlying the validity of the logarithmic mean temperature difference usually employed in heat exchanger design. The effect can be shown to be not serious, and it is actually very slightly conservative to use the conventional F-LMTD formulation.

The coefficient as given by Eq. (3.33) is useful if one knows the condenser tube length, but is awkward if one is trying to design a condenser for a given duty. The equation can be reworked to a more convenient form if one first defines a tube loading per linear foot of tube drainage perimeter \( \Gamma \). That is, if \( w \) lb/hr are to be condensed on each tube,

\[
\Gamma = \frac{w}{P_t}
\]  

(3.34)

where \( P_t = \pi d \) for a vertical tube. The total heat load per tube is

\[
Q = \lambda w = h_c \pi d L (T_{\text{sat}} - T_w)
\]  

(3.35)

With some rearrangement, we obtain the desired equation:

\[
h_c = 0.924 \left[ \frac{k_f \rho_f (\rho_f - \rho_v) g}{\mu_f \Gamma} \right]^{1/3}
\]  

(3.36)

For later purposes, it is also desirable to define a condensate Reynolds number

\[
\text{Re}_c = \frac{4 \Gamma}{\mu_f}
\]  

(3.37)

Substituting this into Eq. (3.36) gives

\[
h_c = 1.47 \left[ \frac{k_f \rho_f (\rho_f - \rho_v) g}{\mu_f^2} \right]^{1/3} \text{Re}_c^{-1/3}
\]  

(3.38)

Before developing the treatment further, it is useful to re-examine the several assumptions of the Nusselt derivation. In general, the validity of the Nusselt equation has been established in experiments in which care has been taken to satisfy the assumptions. But what if those assumptions are not satisfied in an actual application? Does a departure from ideality completely invalidate the equation, or can the equation or its application be modified to still give useful design results? Only by considering each of the assumptions can we answer that.
The consequences of the violation of Nusselt's basic assumption, i.e., laminar flow of the condensate film, are very significant and will be examined at greater length later. Within this assumption, however, many workers have analytically and experimentally tested the effect of violation of the other assumptions listed above. We will now consider these briefly, with attention focused upon the design consequences of the result.

The first assumption, that of saturated vapor, has been studied experimentally and theoretically; the weight of the evidence is that superheating effects in pure vapors are small and Nusselt's equation can be safely applied, if the sensible heat load required to desuperheat the vapor to saturation is added to the latent heat load in calculating condenser duty and if the mean temperature difference for the condenser is calculated using the vapor saturation temperature. This, of course, assumes that the condenser surface temperature is below saturation, so that condensation does occur. The method of calculating surface temperatures will be considered when desuperheating in condensers is discussed.

The second assumption, that of no interfacial resistance, has been a subject of investigation in many areas of transport processes. The effect of an interfacial resistance becomes significant only at extremely high transfer rates for condensation processes; for example, interfacial resistance has been suspected as the phenomenon responsible for liquid metal condensation coefficients being “only” about 10,000 BTU/hr ft²°F, instead of the 100,000 or so predicted by Nusselt's equation. For air cooled condensers, it is completely immaterial whether the assumption is true or not.

The third assumption, concerning the mechanism of heat transfer in the liquid film is exactly as valid as the assumption of laminar flow and breaks down only when the flow is no longer laminar.

The fourth assumption, that of a linear temperature profile in the film, can be relaxed by replacing $\lambda$ by

$$\lambda \left[ 1 + \frac{0.68\lambda}{C_{p,l}(T_{sat} - T_w)} \right]$$

This is a significant correction only when $(T_{sat} - T_w)$ is relatively quite large, usually out of the range of process practice. Since the effect is to increase the calculated value of $h$, it is generally conservative to neglect this correction.

The fifth assumption, equilibrium at the liquid-solid interface, has not been questioned on theoretical grounds, but in practical cases, allowance must be made for a dirt film resistance to heat transfer. This is handled in the conventional way and does not affect the Nusselt equations.

The sixth assumption, an isothermal condensing surface, is generally not realized in practice. While the remaining features of the Nusselt analysis can be rigorously applied to a non-isothermal surface, the calculation requires a computer, and standard practice is to simply use the average computed surface temperature in a condenser. Even this in principle requires a reiterative calculation; assuming a surface temperature, calculating the coefficient, checking the surface temperature, etc. For the usual process case, the effect of an assumption of a mean wall temperature can be shown to be small and probably conservative.

The seventh assumption, constant liquid properties, is never strictly valid, but is accepted because the alternatives are so formidable. The physical properties are generally taken at the arithmetic mean film temperature. Significant errors in the final result are unlikely to arise unless the temperature difference is
very great or the condensate has a very large temperature coefficient of viscosity. In case of doubt, use the viscosity at the surface temperature in the Nusselt equation.

The assumption concerning vapor shear on the condensate on a vertical surface was examined by Nusselt himself (7) and is described in detail in Jakob (10). If the vapor and the condensate flow together vertically downwards, vapor shear somewhat enhances the condensing coefficient in laminar flow (the role of vapor shear in causing laminar flow to be destroyed is discussed later.) If the vapor and condensate flow in opposite directions, the condensate film is thickened and resistance increases; however, in this case a probable and significant consequence is that the film becomes rippled and/or turbulent, and entrainment, slugging, flooding, etc., occur.

Assumption 9, no slip at the liquid-solid interface, is very strongly supported by many studies on laminar flow.

The last assumption, that the heat load from subcooling the liquid is negligible compared to the latent heat, while generally true, can be easily and conservatively relaxed by adding this amount to the total heat to be transferred. The condensate film is subcooled (on the average) to \( T_{sc} = \frac{3}{8} (T_{sat} - T_w) \), so the added heat load is

\[
Q_{sc} = \frac{3}{8} w c_p, T_{sat} - T_w
\]

(3.40)

Possibly a more important and interesting point is that this degree of subcooling may be sufficient (if reheating is avoided) to satisfy the NPSH requirements of the condensate pump and eliminate the need for a condensate subcooler.

Turbulent Film Condensation in Vertical Tubes. Since Eqs. (3.33) and (3.38) are valid only for laminar flow of the condensate film in the vertical tube, we now consider the following questions:

1. When does the film cease to be laminar?
2. What correlation is valid when the film becomes turbulent?
3. What is the effect of vapor shear on the condensation mechanisms and correlations?

With reference to the first question, there is no hard and fast answer to the criterion for transition from laminar to turbulent flow. Ripples can appear on the surface at quite low values of Re_c (as diagrammed in Fig. 3.8) but these seem to have very little effect on the condensing coefficient. There is a definite break in the heat transfer behavior of films at Re_c of 1600 to 2000 in the absence of vapor shear, and this number can be used here as the critical Reynolds number for the falling film.
The presence of vapor shear causes an early appearance of turbulence, \( \text{Re}_c \) values as low as 250 to 300 being reported by Carpenter and Colburn (12). As it turns out, from a practical point of view, the question is not a serious one since the magnitude of the calculated coefficient itself indicates what flow regime exists.

Condensation heat transfer coefficients under turbulent flow but low vapor shear conditions are correlated by a fundamental analysis due to Colburn (13) based on analogies between heat and momentum transfer in single phase turbulent flows. The Colburn approach required a numerical integration to get the mean coefficient so that a graphical representation of the final result is usually presented. For this purpose, it is convenient to note that the Nusselt solution can be plotted as

\[
\frac{h}{k} \left[ \frac{l^2}{k_l \rho_l (\rho_l - \rho_v) g} \right]^{1/3} \quad \text{vs} \quad \text{Re}_c
\]

with a slope of -1/3 on log-log coordinates and an intercept of 1.47 at \( \text{Re}_c = 1 \). The Colburn solution may also be plotted on these coordinates with \( \text{Pr}_f \) as a parameter, giving the graph shown here as Fig. 3.9. The break at \( \text{Re}_c = 2100 \) shows the point at which the film is presumed to become turbulent; Colburn used this value in his computation, and it is somewhat conservative. Notice also that the Colburn curves have a Prandtl number dependence. The correlation shown in Fig. 3.9 has been reasonably well verified experimentally, but it seems unwise to extrapolate the Prandtl number dependence to values higher than 5; available data do not go very much higher. (High Prandtl numbers usually arise from higher viscosity, which also causes lower Reynolds numbers, so the problem is in some sense self-limiting. However, the physical structure of a high-flow-rate, high viscosity condensate film may be a thick laminar layer close to the cold wall with a turbulent liquid film cascading down the outside. This could result in a higher heat transfer resistance than expected from a condensate layer whose properties are calculated at a mean temperature.)

![Graph](image)

Fig. 3.9 Correlation for Condensation on a Vertical Surface—No Vapor Shear.

Turning now to the case of a vapor-shear controlled condensing situation inside a vertical tube, there are several procedures to choose from in the literature. Carpenter and Colburn (12) did the first really comprehensive experimental study of this problem, correlating their average coefficients by
\[
\frac{h_c \mu \ell}{k_e \rho \ell^{1/2}} = 0.065 \quad \text{Pr}_\ell^{1/2} \quad F_{vc}^{1/2}
\]

where

\[
F_{vc} = \frac{fG_{v,m}^2}{2\rho_v}
\]

(3.42)

\[
G_{v,m} = \left( \frac{G_{v,d}^2 + G_{v,i}G_{v,o} + G_{v,o}^2}{3} \right)^{1/2}
\]

(3.43)

and \(f\) is the Fanning friction factor, given by

\[
f = \frac{0.078}{\left( \frac{d_j G_{v,m}}{\mu_v} \right)^{1/4}}
\]

(3.44)

\(G_{v,i}\) and \(G_{v,o}\) are the vapor phase mass velocities at the inlet and outlet, respectively. If the vapor comes in dry and saturated and is completely condensed, \(G_{v,m} = 0.58 G_{v,i}\).

There are numerous other correlations for condensation in the presence of high vapor shear, and some of them are more accurate. However, the more accurate ones are also harder to use, and usually the additional accuracy is not required in air-cooled condensers, where the condensation is not the controlling resistance.

The Carpenter-Colburn correlation is valid only under the condition that vapor shear controls the liquid film hydrodynamics and hence heat transfer. If vapor shear does not control, this correlation will give an unrealistically low coefficient. Therefore, in deciding which correlation to use, the following rule applies: Calculate condensing coefficients by the gravity-controlled correlation (Eq. 3.36) and by the vapor shear-controlled correlation (Eqs. 3.41 to 3.44) and take the higher value.

**Laminar Film Condensation Inside Horizontal Tubes.** Essentially the same set of assumptions as previously used for vertical tubes may be applied to condensation in a horizontal tube. The only essential difference between the analyses for the two cases is that:

a. for the horizontal tube, the effective component of the gravitational acceleration, \(g \sin \theta\), changes with position around the tube, and

b. the condensate will form a pool in the bottom of the tube (Fig. 3.10) and render that part of the tube surface ineffective.

**Fig. 3.10 Nusselt Condensation Inside a Horizontal Tube**
The fraction of surface thus affected depends upon the properties of the condensate, the rate of condensation, the geometry of the tube, and the provision made for removing the condensate. If there were no condensate pool, Nusselt's analysis gives the heat transfer coefficient as Eq. (3.45)

\[
h_c = 0.725 \left[ \frac{k^3 \rho_f (\rho_f - \rho_v) g}{\mu_f d_f (T_{sat} - T_w)} \right]^{1/4}
\]

(3.45)

Kern (11) recast this equation in a form that introduces the condensate weight flow rate per tube \(W_t\), eliminates the temperature difference, and introduces a penalty for the presence of the pool. His equation is:

\[
h_c = 0.761 \left[ \frac{k^3 \rho_f (\rho_f - \rho_v) g L}{W_t \mu_f} \right]^{1/3}
\]

(3.46)

Again, more elegant equations are available, but the presumed additional precision is not essential. Eq. (3.46) is valid only at low vapor shear rates.

At high condensing loads, with vapor shear dominating, the correlations should be independent of tube orientation, and this is in fact found to be the case. Therefore, the high-vapor-shear condensing coefficient may be calculated from the Carpenter-Colburn equations (Eqs. 3.41 to 3.44) previously given for this case.

The selection of which correlation to apply in a given case is based on the following argument: The flow patterns, the hydrodynamics and consequently, the heat transfer processes are dominated by gravity at low tube loadings and by vapor shear at high loadings. Correlations are available for both limiting cases; each of these correlations predicts low coefficients (relative to the appropriate correlation) when it is applied to situations to which it is in fact not applicable. Therefore, in any case in which there is doubt as to the correct correlation, calculate the condensing coefficient by each correlation (Eq. (3.41 to 3.44) and Eq. (3.46)) and choose the higher value of \(h_c\).

**Filmwise Condensation Inside Inclined Tubes.** In the foregoing discussion of condensation inside vertical and horizontal tubes, the argument was made that tube orientation was unimportant at high velocities where vapor shear controlled, and we would expect the same to be true of condensation in inclined tubes with vapor flow towards the lower end of the tube. In the gravity-controlled regime, however, inclination should make a difference. Very little information is available in the open literature on this problem and the best we can do here is suggest a procedure that will be responsive to the major effects and reduce to the correct limiting cases.

If we start with a vertical tube \(\theta = 0\), as defined in Fig. 3.6), we would expect an increase in \(\theta\) to reduce the effective component of the gravitational force acting on the draining film by the function \(g \cos \theta\). Therefore, it is suggested that the condensing heat transfer coefficient for an inclined tube in the gravity-controlled regime be calculated from Fig. 3.9, using as the ordinate
instead of that shown.

But if that procedure is carried to nearly horizontal tubes, where \( \theta \to 90^\circ \), \( \cos \theta \to 0 \) and \( h_c \to 0 \), which is physically unrealistic. It hardly seems likely that \( h_c \) would fall below the value for a horizontal tube; indeed, even a few degrees of downward slope should aid greatly in the drainage and improve the coefficient above the horizontal case. Therefore, it seems conservative to place the lower bound on \( h_c \) (of an inclined tube) as the value given by Eq. (3.46) for a horizontal tube at the same loading.

Additionally, in each case, the value of \( h_c \) predicted by Eqs. (3.41 to 3.44) should be calculated. If the value so calculated is greater than that obtained in the previous paragraph, the flow may be presumed to be predominantly in the vapor shear-controlled regime and the higher value of \( h_c \) used for further calculations.

### 3.2.3. Mean Temperature Difference for In-Tube Condensation

**MTD for a Pure Saturated Vapor.** For the usual conditions of condensing a pure saturated vapor in an air cooled exchanger, the correct value of the Mean Temperature Difference (MTD) is the logarithmic mean temperature difference, using the saturation temperature, \( T_{\text{sat}} \), of the vapor at the nominal condensing pressure as the constant hot side temperature:

\[
MTD = \frac{T_0 - T_i}{\ln \left( \frac{T_{\text{sat}} - T_i}{T_{\text{sat}} - T_0} \right)}
\]  
\[
(3.47)
\]

Strictly speaking, this equation is only valid if the overall coefficient is constant and if the condensing fluid temperature is isothermal, i.e., if there is no desuperheating or subcooling and the vapor pressure drop is very small compared to the absolute pressure. In fact, these are usually not serious limitations. Since the condensation process is commonly only a relatively small part of the total resistance to heat transfer, the overall coefficient varies little about a mean value calculated on the basis of an average condensing coefficient, or for some correlations, a condensing coefficient calculated at the average quality of the condensing stream. The treatment of desuperheating and subcooling is given later. The other condition - negligible pressure drop in the condensing process may be relaxed approximately but satisfactorily by finding the saturation temperatures at the inlet and outlet pressures, \( T_{\text{sat},i} \) and \( T_{\text{sat},o} \), calculating the LMTD as

\[
LMTD = \frac{(T_{\text{sat},o} - T_i) - (T_{\text{sat},i} - T_0)}{\ln \left( \frac{T_{\text{sat}} - T_i}{T_{\text{sat}} - T_0} \right)}
\]  
\[
(3.48)
\]

and finally calculating the mean temperature difference as

\[
MTD = F(LMTD)
\]  
\[
(3.49)
\]
where \( F \) is found from the curves in Chapter 4. This is a reasonably valid procedure only if the temperature approach is not too close and if the pressure drop in the condensing vapor is relatively small. Both of these conditions are usually met in air-cooled condensers.

Most air-cooled condensers are designed so that the condensation takes place in a single pass from one header to the other. If this is not done, the liquid and vapor phases will tend to separate and mal-distribute in the turn around header, resulting in some tubes having a surplus of vapor (and, therefore, possibly not giving complete condensation) and other tubes having a surplus of liquid (and giving excessive subcooling.)

However, it is usually necessary for the single condensing pass to include several vertical tube rows. Since the air becomes progressively hotter from row to row, the local temperature difference and the condensation rate correspondingly decrease. This means that if the vapor in the bottom row of tubes is just completely condensed (no subcooling), the vapor in the upper rows is in completely condensed, leading to a loss of vapor.

In order to avoid this, Mueller (14) has derived a safety factor to be applied to assure that the vapor entering the top rows is completely condensed by the end of the tube (resulting in some subcooling of the condensate in the lower rows). The factor is

\[
E = \frac{\text{Total tube length in exchanger}}{\text{Total tube length required for condensation}} = \frac{n + n(n-1)K}{n + \frac{n(n-1)K}{2}}
\]  

(3.50)

where

\[
K = 1 - \exp\left[-\frac{UA}{W_{\text{air}}C_{\text{air}}}\right]
\]  

(3.51)

and where \( U \) is the overall heat transfer coefficient, based on the total outside heat transfer area \( A \) (actually, any consistent combination of \( U \) and \( A \) may be used) and \( N \) is the number of rows of tubes in the exchanger. The heat transfer area (and therefore the tube length) required for complete condensation is calculated assuming a uniform condensing loading for each row of tubes, and is then multiplied by \( E \) from Eq. (3.50) to find the total area (and the additional tube length) required to assure complete condensation in the top row.

Desuperheating of vapor. If the vapor entering a condenser is superheated, the sensible heat content of the vapor must be removed and transferred through the cooling surface before that vapor can be condensed. If the cold surface is above the saturation temperature of the vapor, the heat is removed by a convective sensible heat transfer mechanism, the coefficient for which can be calculated from correlations given in Chapter 2 employing vapor physical properties. However, if the cold surface is below the saturation temperature of the vapor at the existing pressure, vapor will condense directly upon the surface with essential thermodynamic equilibrium existing at the condensate-vapor interface and with the temperature gradient from the superheated state to saturation occurring in the vapor immediately adjacent to the interface.

Available information indicates that the heat transfer coefficient for condensation directly from the superheated vapor is within a few percent of that for condensation from the saturated vapor, using in each case the saturation temperature of the vapor as the temperature driving force for heat transfer. Such
differences as have been observed are well within the present ability to predict condensing coefficients for a given situation. This fortuitous agreement is useful in the design of desuperheating condensers.

The first matter is to establish a test to determine whether or not condensation will occur on a cold surface exposed to superheated vapor. If we assume that heat is transferred from the superheated vapor by sensible heat transfer, and if \( T_v \) is the local vapor temperature, \( T_{\text{sat}} \) the condensing or saturation temperature, and \( t \) the local coolant temperature, the wall temperature on the vapor side, \( T'_w \), is given by

\[
T'_w = T_v - \frac{U_s(T_v - t)}{h_s}
\]  

(3.52)

where \( h_s \) is the sensible heat transfer coefficient for the vapor stream and \( U_s \) is the overall coefficient computed using \( h_s \), both referenced to the same surface. If \( T'_w > T_{\text{sat}} \), no condensation will occur. \( T'_w \) is equal to the true wall surface temperature \( T_w \) and the heat transfer rate is given by \( (Q/A) = U_s(T_v - t) \). If \( T'_w < T_{\text{sat}} \), condensation will occur and the heat transfer rate is given by \( (Q/A) = U_c(T_{\text{sat}} - t) \), where \( U_c \) is the overall coefficient computed assuming condensation does occur, and the true inside tube surface temperature is given by:

\[
T_w = T_{\text{sat}} - \frac{U_c(T_{\text{sat}} - t)}{h_c}
\]  

(3.53)

There is a further interesting consequence of the above discussion. Let \( U^* \) be the combined heat transfer coefficient for the wall and fin resistance, the coolant and any dirt films. \( U^* \) is essentially independent of the heat flux and whether or not condensation is occurring inside the tube. Then, the heat flux for the sensible heat transfer desuperheating case is \( (Q/A) = U^*(T'_w - t) \) and that for the condensing case is \( (Q/A) = U^*(T_w - t) \). Since \( T'_w \geq T_w \), we conclude that condensation will occur directly from the superheated vapor, unless a higher heat flux is obtained by the sensible heat transfer mode. A corollary to this is that it is both simpler and more conservative (in the sense of calculating a larger condenser area) to assume that condensation will occur directly from the superheated vapor, using the saturation temperature and a condensing heat transfer coefficient in the rate equation, and of course, including the sensible heat in the heat load.

However, in designing a desuperheating condenser where the desuperheating heat load is an appreciable fraction of the total heat load, the designer may wish to avail himself of the possible surface savings afforded by the higher heat flux of sensible desuperheating. This is a fairly complicated matter, since the heat balance and rate equations must be simultaneously balanced by increments in two directions - vertically through the tube bundle and longitudinally along the length of the tubes. This is strictly a computer solution; proprietary programs are available and they run quite rapidly in both the rating and design modes. It must be remembered that the computer solution is only as good as the data base and the skill of the engineer and systems analyst who put the program together, and even the best of the methods have a substantial, range of uncertainty in the final answer.

\textit{Subcooling the Condensate.} It is often desirable to further subcool the condensate beyond the few degrees achieved by the condensing process itself. This can be done by passing the condensate through one or more rows of tubes in the bottom of the air cooler.

Again, the exact thermal analysis of this problem is quite complicated and can only be carried out by a computer program. An \textit{approximate} calculation can be carried out by considering the condensing and subcooling heat transfer processes to be carried out in two separate heat exchangers in series. Thus one can calculate the mixed mean air temperature just after passing over the subcooling tubes, using the
methods outlined in Chapter 4 on air cooling. This mixed mean air temperature can then be used as the inlet air temperature to the condensing section, which is designed according to the methods in this chapter.

There are several important cautions to be observed. First, the air off of the subcooling sections varies along the length of the tube, and this variation alters the condensing pattern, increasing the local temperature difference (compared to the mixed mean assumption) and local condensation rate at one end and decreasing it at the other. The effect tends to cancel out, though whether the net effect is conservative or non-conservative depends upon the specifics of each problem. Certainly the absolute error of the method increases as the process stream temperatures approach more closely the air temperatures.

Another consideration is this: Almost certainly the design and the mechanical layout of the air-cooled exchanger will be dominated by the condensing process. Therefore, the subcooling achieved will be at the mercy of the conditions set for the condensing process and not independently controllable. If close condensate subcooling control is required, it is much better to provide a separate heat exchanger, the operating conditions for which can be adjusted as required.
3.3. Condensation of Vapor Outside Low- and Medium-Finned Trufin Tubes

This section is concerned with the application of low- and medium-finned Trufin tubes to cases where condensation (including desuperheating and subcooling) is taking place on the finned surface of the tubes. Typical applications for outside condensing include:

Condensing refrigerants in air conditioning, refrigeration and cryogenic processing systems with cooling water. Typical refrigerants have heat transfer coefficients from about 150 to 500 Btu/hrft²°F; the water coefficient will usually be from 1000 to 1500, so the advantages of using Trufin on the condensing side are clear. Condensing overhead streams for distillation columns in water-cooled condensers. An overhead stream from a distillation column may be composed of essentially one component or it may contain many components. In either case, the condensing coefficient is likely to be less than half of the water coefficient. If the condensing temperature range is very long (for a multi-component mixture, or for a vapor containing a non-condensable gas), a large portion of the total heat transfer may be sensible cooling of the remaining gas or vapor. This coefficient is almost always very low compared to the coolant coefficient, or even the condensing coefficient, making the use of Trufin even more attractive.

Condensing vapor using a non-aqueous coolant. In this case, the coefficients of the two fluids may be nearly equal and there might seem to be little advantage in using standard Trufin. However, a doubly enhanced tube such as Turbo-Chil has internal spiral ridges as well as conventional external fins. These ridges enhance the tube side coefficient and the combined enhancement can lead to a very substantial reduction in heat exchanger size.

One application for which Trufin is not recommended is the condensation of water vapor (steam). Because of its high surface tension, water tends to bridge the gaps between the fins and a thick layer is retained on the tube, severely reducing the effective heat transfer coefficient. For this application, Korodense can be advantageous.

3.3.1. Shell and Tube Heat Exchangers for Condensing Applications

Shell and tube exchangers offer a mechanically feasible way of providing a large heat transfer area in a relatively compact volume. The configuration is strong enough to withstand a wide range of operating conditions commonly encountered in the process and power industries, and offers an enormous number and range of design options to meet most requirements.

The basic construction of the shell and tube exchanger is described in Chapter 1, and we will not repeat that material here. However, there are several special configurations of shell and tube exchanger that are particularly interesting in condensing applications, and these are described in the following paragraphs (15).

Fig. 3.11 shows a typical shell and tube condenser as employed in the process industries. Condensation occurs on the shell-side, the vapor entering through a nozzle at one end and the condensate being removed from the nozzle on the bottom side of the shell at the other. There is also a vent for non-condensable gases at the condensate exit end of the condenser. This configuration is known, by TEMA notation, as an E shell. The drawing shows a fixed tube sheet, two coolant pass arrangement. The baffling is shown in Section AA in the most usual configuration, that is, segmentally cut baffles with a vertical cut and the baffles notched on the bottom to allow drainage of the condensate from one compartment to the next and finally to the condensate exit. However, there has been a recent trend to
use horizontally-cut baffles so that the two-phase flow must go up and over; this arrangement is believed to minimize the possibility of stratification and liquid segregation on the shell-side, but at the cost of additional pressure drop. An impingement plate is shown at the vapor inlet to protect the tubes immediately adjacent to the vapor nozzle from erosion from liquid droplets carried along in the vapor. It is almost universal practice to use an impingement plate at the vapor inlet on a shell-side condenser.

Fig. 3.11  Shell and Tube Exchangers Arranged for Shell-Side Condensation. One Shell-Side Pass, Two Tube-Side Passes.

It is often necessary in heat recovery service (e.g., feed effluent exchangers) to carry out condensation in several exchangers in series. Shell-side condensation in stacked horizontal E shells is the usually preferred design in this case (Fig. 3.12). This arrangement minimizes (but does not eliminate) the possibility of phase segregation and the resultant distortion of the multicomponent vapor-liquid equilibrium profile. It is still important to select a baffle spacing and cut to keep vapor velocities fairly high, and the baffle geometry will often be different for each shell.

Fig. 3.13 shows a somewhat different configuration for shell-side condensation, referred to in the TEMA notation as a G shell. The flow is split into two streams which proceed more or less symmetrically from the center vapor inlet to the ends of the tubes, being guided in this direction by the longitudinal baffle. The flow is then turned around in the end baffle sections and brought back through the lower portion of the tube field; the condensate exits from the central hot well of the condenser. Again, the standard practice is to make the baffle cuts vertical so that the flow is from side to side.

Fig. 3.14 shows the TEMA J shell configured as a condenser. The baffle cuts are usually vertical. Dividing the flow and halving the flow distance (compared to the corresponding E shell) results
in a reduction of 60 to 80 percent in pressure drop, a consideration of especial importance in vacuum service. However, the temperature profiles require careful analysis because they are not identical for the two sections.

J shells can be stacked in series as shown in Fig. 3.15. As drawn, the coolant flow is in parallel to the two shells; this maximizes the effective temperature difference for condensing but can only be used when abundant coolant is available.

Fig. 3.16 shows a further modification of shell-side condensing arrangements, commonly referred to as an H shell. This is effectively a double G shell arrangement, commonly, called a double split flow, and is used to reduce the pressure drop. These units are frequently used for vacuum condensing applications. The transverse baffles have vertical cuts and overlap only enough to insure tube support, i.e., the overlap will ordinarily be two rows of tubes at the vertical centerline of the shell. The configuration shown uses a U tube bundle in order to show possible variant design configurations; however, fixed tube sheet/single tube-side pass and other design options are feasible with this particular shell-side geometry.

If pressure drop is extremely limited, the usual design choice will be the X shell shown in Fig. 3.17. This arrangement provides that the vapor flow is essentially in cross-flow with the tube bank and is always arranged so that gravity will remove the condensate from the tube surfaces. There must be sufficient clearance between the top of the baffles and the shell to allow the vapor to disperse longitudinally across the entire length of the tubes. Sometimes this is accomplished by putting a large longitudinal vapor nozzle on top of the shell; this arrangement is descriptively referred to as a bathtub nozzle. If adequate space is provided between the tube field and the shell for longitudinal flow of the vapor, each transverse baffle can be a full baffle giving full tube support to the entire bundle. The baffles exist only for tube support, and their spacing is controlled by vibration requirements. The arrangement shown here has a four coolant pass arrangement. The vent for this design would ordinarily be located in the condensate hot well, which would be designed to maintain a vapor/liquid interface below the tube field. The non-condensables would tend to accumulate at the top of the hot well and could be vented from there. Vents at the far end of the
shell would also usually be required in order to permit removal of any non-condensable gases that might otherwise accumulate there and not be swept down to the hot well.

Another design to meet a special need is the shell-side reflux or knock-back condenser shown in Fig. 3.18. A typical application is to partially condense a wide condensing range vapor-gas mixture. The first liquid to condense is tarry but will dissolve in the lighter liquid condensed near the top of the bundle. By
keeping the gas/vapor velocity low (an essential in all reflux condensers), the lighter liquid drains downwards constantly washing off the tarry material.

Fig. 3.19 shows a recent proprietary innovation in shell side condensation in the rod baffle design, licensed by Phillips Petroleum Company (16). In this case, each segmental baffle is replaced by a set of four individual baffles, each one composed of a ring that fits close to but inside the shell, with straight rods extending from one side of the ring baffle to the other. The straight rods are so arranged that they pass between the tube rows with minimum clearance. Every point at which a rod passes next to a tube provides a point of support for that tube in that direction. The rods are arranged so that there are two rows of tubes between each pair of rods in the baffle; the next baffle in line has offset rods passing between the two tube rows that were not individually supported by the previous baffle on that side. The individual baffles are spaced 6 to 8 in. apart, i.e., each tube being supported on all four sides every 24 to 32 in. along the length of the tube. This configuration was first introduced to prevent vibration in an extremely high velocity service but it is now recognized to be at least as valuable for condenser application because of the low pressure drop caused by this particular selection of rod geometry.

Some other shell-side condensing options have considerable application. Fig. 3.20 shows variable baffle spacing on the condensing side. The intention here is to maintain high vapor velocity and therefore shear enhancement of the condensing heat transfer coefficient. Clearly, a fair amount of knowledge on the effect of vapor shear and the ability to predict two-phase flow characteristics on the shell-side is required if this option is to be effectively exercised. This arrangement can be used only when there is a reasonable amount of pressure drop allowed on the condensing side to permit maintaining vapor velocities at a high enough value to get shear enhancement. Removal of non-condensable gases in this geometry is quite positive as long as a vent is located on the exit end of the shell, because this is where the natural flow of the vapors will carry the gas. A computer based rating method is essential to the effective use of this option, and detailed discussion of such applications is beyond the scope of this Manual.
Fig. 3.21 shows a vapor belt distributor, which is intended to uniformly distribute the vapor around the bundle and therefore reduce the excessively large pressure drops and vibration and erosion problems encountered at the vapor inlet end. The shell is extended underneath the vapor nozzle thereby acting as an impingement baffle. Usually, the extended shell within the vapor belt distributor has slots cut around the periphery in order to introduce the vapor into the tube field from all angles. In this case, the open end of the shell shown in the diagram is not needed. The vapor belt distributor is a relatively high-cost construction option but is particularly useful where pressure drop may be a critical consideration.

All vapors introduced into a condenser will contain some amount of non-condensable gas, at least during certain portions of the operational period, whether this composition is shown in the process specifications or not. This fact gives rise to the basic rules for venting condensers: 1. All condensers must be vented. 2. Condensers must be designed to move the non-condensables to a particular point in the condenser by directing the vapor flow path positively through the use of baffles and other mechanical arrangements. 3. The vent must be located where the non-condensables are finally concentrated. 4. A vent condenser may be necessary, sometimes even using a refrigerated coolant, in order to recover as much of the remaining condensable vapor from the non-condensable gas as possible.

It is fair to say that probably half of all operational condenser problems arise from a failure to recognize the basic principles of venting non-condensable gases.

In condensation applications it is frequently necessary to subcool the condensate produced in the condenser. The required amount of subcooling may vary from a few degrees to prevent cavitation in a pump to many degrees required for cooling the condensate to a safe temperature for long-term atmospheric storage. The surest way to carry out the subcooling, especially if large amounts of subcooling are required, is to provide a separate subcooler as shown in Fig. 3.22. In this case, the subcooler is designed as a single-phase heat exchanger with far more precise design methods than are available for integral subcooling calculations. Additionally, the coolant flow rate may be separately controlled to the subcooler giving better control of effluent liquid temperatures. The obvious disadvantage is the generally higher first cost of the separate condenser and subcooler, compared to the cost of the condenser with integral subcooling.

Integral subcooling can be accomplished in a horizontal shell-side condenser by allowing a stratified pool to accumulate in the shell and be sensibly cooled by the inlet coolant, as shown in Fig. 3.23. Some sort of level control (preferably adjustable) is required, and the design calculations are both fairly complex and not very accurate.
3.3.2. The Basic Design Equations

The basic heat exchanger equations applicable to shell and tube exchangers were developed in Chapter 1, and the reader is referred to that material for the development. Here, we will cite only those that are immediately useful for design in shell and tube heat exchangers with condensing heat transfer on the shell side. Specifically, for the moment, we will limit ourselves to the case when the overall heat transfer coefficient is constant and the other assumptions of the mean temperature difference concept apply. (The important cases of condensation of multi-component vapor or a vapor with a non-condensable gas do not satisfy this requirement and are discussed later.) The basic design equation becomes:

$$Q_T = U^* A^* (LMTD)$$

(3.54)

where $Q_T$ is the total heat load to be transferred, $U^*$ is the overall heat transfer coefficient referred to the area $A^*$. $A^*$ is any convenient heat transfer area, and LMTD is the logarithmic mean temperature difference.

$U^*$ is most commonly referred to as $A_o$, the total outside tube heat transfer area, including fins, in which case it is written as $U_o$ and is related to the individual film coefficients, wall resistance, etc. by

$$U_o = \frac{1}{\frac{1}{h_o} + R_{f_o} + \frac{\Delta h_w}{h_w} \left( \frac{A_f}{A_w} \right) + R_w \left( \frac{A_w}{A_t} \right) + \frac{1}{h_t} \left( \frac{A_t}{A_f} \right)}$$

(3.55)

where $h_o$ and $h_t$ are the outside and inside film heat transfer coefficients, respectively, $R_{f_o}$ and $R_w$ are the outside and inside fouling resistances, $\Delta h_w$ and $k_w$ are the wall thickness (in the finned section) and wall
thermal conductivity, and $R_{m}$ is the resistance to heat transfer due to the presence of the fin. Since all of the low and medium Trufin tubes manufactured by Wolverine are integral (i.e., tube and fins are all one piece of metal), there is no need to include a contact resistance term.

Suitable correlations for $h_{l}$ will be developed in this section. Correlations for $h_{i}$ have been developed in Chapter 2. Equations and charts for fin resistance and Wall resistance calculations are found in Chapter 1.

3.3.3. Mean Temperature Difference

The Mean Temperature Difference formulation is detailed in Chapter I but must be used very carefully in condensation applications. For the special but important case of condensation of a pure or nearly pure vapor with negligible pressure drop on the condensing side (i.e., isothermal condensation), the Logarithmic Mean Temperature Difference is not only valid but considerably simplified. If the condensing temperature is $T_{\text{sat}}$ and the inlet and outlet coolant temperature are $t_{1}$ and $t_{2}$, then the LMTD (and the MTD) is given by:

$$LMTD = \frac{t_{2} - t_{1}}{\ln \left( \frac{T_{\text{sat}} - t_{1}}{T_{\text{sat}} - t_{2}} \right)}$$  (3.56)

(Strictly speaking, since most condensing heat transfer correlations are functions of the local heat flux or the local temperature difference, the LMTD is not exactly valid. However, for practical cases, the effect is small and the use of the LMTD is generally conservative by engineering practice).

If multi-component mixtures or vapors with non-condensable gases are being condensed, or if there are appreciable desuperheating or subcooling effects, the MTD/LMTD formulation is not valid. Special techniques need to be used for these cases; these are described later.

3.3.4. Condensation of a Superheated Vapor

It is often necessary to condense a vapor whose inlet temperature is above the saturation temperature at the pressure existing in the condenser. Such a case can arise with exhaust vapor from a non-condensing turbine or a vapor throttled through a valve. There is some confusion in the literature about the proper way to design for condensing a superheated vapor, but the following argument [shortened from Ref. (17)] is generally accepted as correct.

If the vapor entering a condenser is superheated, the sensible heat content of the vapor must be removed and transferred through the cooling surface before that vapor can be condensed. If the cold surface is above the saturation temperature of the vapor, the heat is removed by a convective sensible heat transfer mechanism, the coefficient for which can be calculated from a correlation applicable to the geometry involved and employing vapor physical properties. However, if the cold surface is below the saturation temperature of the vapor at the existing pressure, vapor will condense directly upon the surface with essential thermodynamic equilibrium existing at the condensate-vapor interface and with the temperature gradient from the superheated state to saturation occurring in the vapor immediately adjacent to the interface.

The information available [Refs. (18,19)] indicates that the heat transfer coefficient for condensation directly from the superheated vapor is within a few percent of that for condensation from the saturated
vapor, using the saturation temperature of the vapor to calculate the temperature difference for heat transfer.

To determine whether or not condensation will occur on a cold surface exposed to superheated vapor, first assume that heat is transferred from the superheated vapor by sensible heat transfer. If \( T_v \) is the local vapor temperature, \( T_{\text{sat}} \) the condensing or saturation temperature, and \( t \) the local coolant temperature, then the surface temperature on the vapor side, \( T'_s \), is given by

\[
T'_s = T_v - \frac{U_s (T_v - t)}{h_s}
\]  

(3.57)

where \( h_s \) is the sensible heat transfer coefficient for the vapor stream and \( U_s \) is the overall coefficient computed using \( h_s \). If \( T'_s > T_{\text{sat}} \), no condensation will occur, \( T'_s \) is equal to the true surface temperature \( T \), and the heat transfer rate is given by \( (Q/A) = U_s (T_v - t) \). If \( T'_s < T_{\text{sat}} \), condensation will occur and the heat transfer rate is given by \( (Q/A) = U_c (T_{\text{sat}} - t) \), where \( U_c \) is the overall coefficient computed assuming condensation does occur, and the true surface temperature is given by

\[
T_s = T_{\text{sat}} - \frac{U_c (T_{\text{sat}} - t)}{h_c}
\]  

(3.58)

In this case, \( U_c = U_o \) as given by Eq. (3.55), with \( h_c \) in that equation being the same as \( h \) in the above discussion. Correlations for the \( h_c \) are given later.

There is a further consequence. Define \( U' \) as the combined heat transfer coefficient for the wall resistance, the coolant, and any dirt film. \( U' \) is essentially independent of the heat flux and whether or not condensation is occurring on the outer surface. Then the heat flux for the sensible heat transfer desuperheating case is \( (Q/A)_s = U' (T'_s - t) \) and that for the condensing case is \( (Q/A)_c = U' (T_s - t) \). Since \( T'_s > T_{\text{sat}} \geq T_s \), \( (Q/A)_s > (Q/A)_c \). Therefore, condensation will occur directly from the superheated vapor, unless a higher heat flux is obtained by the sensible heat transfer mode. A corollary to this is that it is both simpler and more conservative (in the sense of calculating a larger condenser area) to assume that condensation will occur directly from the superheated vapor, using the saturation temperature and a condensing heat transfer coefficient in the rate equation, and of course including the sensible heat in the heat load.

### 3.3.5. Condensation with Integral Subcooling On the Shellsid

Condensate subcooling is frequently required to provide the NPSH for a pump or to cool a product for a surge tank or for storage. Usually this subcooling is best done in a separate sensibly cooling heat exchanger, specifically designed for this purpose. It should also be remembered that filmwise condensation results in a liquid film that is subcooled on the order of 1/3 to 1/2 of \( T_{\text{sat}} - T_s \). If reheating can be avoided and the pump judiciously located, this subcooling may be sufficient for the purpose.

Occasionally, however, it may be desirable, especially if a moderate degree of subcooling is required, to provide a region in a horizontal condenser shell where the condensate may exchange heat with the cold coolant entering. In this case, a rational approach to calculating the subcooling area required is necessary. Referring to Fig. 3.23, the surface of the liquid pool is shown as essentially horizontal, whereas in reality pressure differences may cause considerable hydrostatic gradient. In this case, the value of \( F \) calculated later must be understood to be the effective fraction of tubes flooded.
In the absence of better knowledge, the pool can be conservatively assumed to be isothermal at $T_{sc}$ the desired subcooling temperature. The condenser therefore may be envisioned as composed of two regions, each isothermal on the shell side: 1) the condensing region at $T_{sat}$ and 2) the subcooling region at $T_{sc}$. The heat load and temperature profile of the coolant as a function of length are then given by

$$Q(x) = wc_p (t-t_i)$$  \hspace{1cm} (3.59)

$$= U_{sc} ax \left( \frac{t-t_i}{T_{sc}-t} \right)$$  \hspace{1cm} (3.60)

in the subcooling region, where $t$ is the temperature of the coolant at any distance $x$ down the tube. Solving for $t$ gives

$$t = T_{sc} - (T_{sc} - t_i) - \frac{U_{sc} ax}{wc_p}$$  \hspace{1cm} (3.61)

At the end of the tube, $x = L$, and $t$ is the outlet temperature for that pass, $t_{bo}$ in the case diagrammed in Fig. 3.23. An analogous equation holds for the condensing region, resulting in a tube outlet temperature of $t_{bo}$ from the tubes in the first pass that are in condensing service. At the turnaround, the two streams from the first pass are presumed to mix completely to give $t_{bo}$ by heat balance, which becomes the inlet temperature for the next pass. In the use of Eq. (3.61) it is immaterial whether $aL$ is taken to be the area per tube and $w$ the coolant flow rate per tube or whether $aL$ is the total area of all tubes in a pass and $w$ the total coolant flow rate. In the following computational scheme, the latter viewpoint is adopted together with the fact that in a two-pass configuration (with equal tubes/pass) $aL = A_T/2$, where $A_T$ is the total tube surface area in the bundle.

The algorithm for this case proceeds by estimating the total area required (step 3) and determining whether the final outlet temperature of the coolant is equal to that required by the heat balance. The total area is then adjusted until the temperatures do match. Since step 3 provides only a very rough estimate, it is possible that the $A_T$ thus predicted will throw the first sequence of calculations into the wrong branch, especially if $F$ (step 5) is near 1. This is readily detected by inspection during the calculations and a more realistic value of $A_T$ can be estimated at steps 9 or 15. The computational sequence is:

1. Calculate the heat loads.
   $$Q_c = W\lambda$$  \hspace{1cm} (3.62)
   $$Q_{sc} = WC_p(T_{sat} - T_{sc})$$  \hspace{1cm} (3.63)

2. Calculate the coolant flow rate.
   $$w = \frac{Q_c + Q_{sc}}{c_p (t_o - t_i)}$$  \hspace{1cm} (3.64)

3. Estimate the total area.
4. Calculate the outlet temperature from the flooded tubes in the first pass.

\[
t_{bo} = T_{sc} - (T_{sc} - t_i) e^{-\frac{U_c A_T}{2 \omega c p}}
\]  

(3.66)

5. Calculate the fraction of tubes in the first pass to be flooded, F.

\[
F = \frac{Q_{sc}}{\omega c p (t_{bo} - t_i)}
\]

(3.67)

If \( F > 1 \), the entire first pass is to be flooded; go to step 10 directly.

6. Calculate the outlet temperature from the unflooded tubes in the first pass.

\[
t_{ao} = T_{sat} - (T_{sat} - t_i) e^{-\frac{U_c A_T}{2 \omega c p}}
\]

(3.68)

7. Calculate the mixed mean outlet temperature from all the first pass tubes.

\[
t_{do} = (1 - F)t_{ao} + Ft_{bo}
\]

(3.69)

8. Calculate the outlet temperature from the second pass (and hence from the condenser.)

\[
t_{o}^* = T_{sat} - (T_{sat} - t_{do}) e^{-\frac{U_c A_T}{2 \omega c p}}
\]

(3.70)

9. Does \( t_{o}^* = t_o \) specified?

Yes: Total condenser – subcooler area required is \( A_T \) and F of the first pass tubes are to be flooded.

No: Assume new \( A_T \) and go to step 4.

10. Calculate the exit temperature from the first-pass tubes, all of which are flooded.

\[
t_{eo} = T_{sc} - (T_{sc} - t_i) e^{-\frac{U_c A_T}{2 \omega c p}}
\]

(3.71)

11. Calculate the exit temperature from the second-pass tubes that are flooded.

\[
t_{fo} = T_{sc} - (T_{sc} - t_{eo}) e^{-\frac{U_c A_T}{2 \omega c p}}
\]

(3.72)
12. Calculate the exit temperature from the second-pass tubes in condensing service.

\[ t_{go} = T_{sat} - (T_{sat} - t_{eo}) e^{-\frac{U_e A_e}{2wcp}} \]  

(3.73)

13. Calculate the fraction of second pass tubes that are in condensing service.

\[ F' = \frac{Q}{wc_p(t_{go} - t_{eo})} \]  

(3.74)

14. Calculate the mixed-mean temperature from the second-pass tubes.

\[ t^* = F't_{go} + (1 - F')t_{eo} \]  

(3.75)

15. Does \( t^*_o = t_o \) specified?

Yes: Total condenser-subcooler area required is \( A_T \) and all but \( F' \) of the second pass tubes are to be flooded, i.e. (1-\( F' \)/2) of all the tubes in both passes are to be flooded.

No: Assume new \( A_T \) and go to step 10.

If U tubes are used in the condenser-subcooler, no mixing of the tube side fluid occurs at the end of the first pass. Assume that the tubes are all the same length and have the same effective heat transfer area; this is not strictly true, but is a convenient assumption and one well within the normal bounds of error for both this case and for other U-tube applications.

The first six steps in the computational scheme are identical with the previous analysis. Then the scheme proceeds as follows:

7. Calculate the outlet temperature from the second pass tubes that were flooded in the first pass.

\[ t_{jo} = T_{sat} - (T_{sat} - t_{bo}) e^{-\frac{U_e A_e}{2wcp}} \]  

(3.76)

8. Calculate the outlet temperature from the second pass tubes that were in condensing service in the first pass.

\[ t_{ko} = T_{sat} - (T_{sat} - t_{ao}) e^{-\frac{U_e A_e}{2wcp}} \]  

(3.77)

9. Calculate the mixed mean outlet temperature from all the second pass- tubes (and hence from the condenser).

\[ t^*_o = F't_{go} + (1 - F')t_{ko} \]  

(3.78)

9A. Does \( t^*_o = t_o \) specified?

Yes: Total condenser-subcooler area is \( A_T \) and \( F \) of the first pass tubes are to be flooded.
No: Assume new $A_T$ and go to step 4.

Steps 10-15 for this case (corresponding to all of the first pass tubes being flooded) are identical to those for the previous case.

### 3.3.6. Filmwise Condensation on Plain and Trufin Tubes

The development of the equations for condensing outside of tubes is based on the same Nusselt model and assumptions that were discussed in detail in the analysis for in-tube condensing. These led to the development of Eq. (3.45) for condensing inside a horizontal tube. By replacing $d_i$ in the equation with $d_o$, the following equation is obtained:

$$h_c = 0.725 \left[ \frac{k_i \rho_i (\rho_i - \rho_v) \lambda g}{\mu_i d_o (T_{sat} - T_w)} \right]^{1/4} \tag{3.79}$$

where $h_c$ is the average condensing heat transfer coefficient on the outside of the tube. Further, $k_i$ and $\mu_i$ are respectively the thermal conductivity and viscosity of the condensate, $\rho_i$ and $\rho_v$, the densities of condensate and vapor respectively, $\lambda$ the latent heat of condensation, $g$ the gravitational acceleration, $d_o$ the outside diameter of the tube, $T_{sat}$ the saturation temperature of the vapor and $T_w$ the wall temperature.

The equation may also be written as:

$$h_c = 0.951 \left[ \frac{k_i \rho_i (\rho_i - \rho_v) g L}{\mu_i W} \right]^{1/3} \tag{3.80}$$

where $L$ is the length of the tube and $W$ is the mass of vapor condensed on the tube per unit time. Other forms of the equation are possible.

These equations are found to predict actual heat transfer coefficients on horizontal single plain round tubes quite closely, on the average about 15 percent lower than the experimental values. The difference is usually attributed to rippling of the film and early turbulence and drainage instabilities on the bottom side of the tube. The equation can also be used with some modification to predict condensing coefficients on Trufin tubes.

Condensation on horizontal low-finned tubes was studied experimentally by Beatty and Katz (20) who found that the data could be correlated by a modified form of Eq. (3.79):

$$h_c = 0.689 \left[ \frac{k_i \rho_i (\rho_i - \rho_v) \lambda g}{\mu_i (T_{sat} - T_w)} \right]^{1/4} \left( \frac{1}{d_{eq}} \right)^{1/4} \tag{3.81}$$

where
\[
\left( \frac{1}{d_{eq}} \right)^{1/4} = 1.3 \Phi \frac{A_{\text{fin}}}{A_{eq}} \left( \frac{1}{L} \right)^{1/4} + \frac{A_{\text{root}}}{A_{eq}} \left( \frac{1}{d_r} \right)^{1/4}
\]

(3.82)

where \( \Phi \) is the fin efficiency, \( A_{\text{fin}} \) is the total fin area, \( A_{eq} \) is the effective outside area of a finned tube, \( A_{\text{root}} \) is the plain tube outside heat transfer area (based on root diameter), and \( d_r \) is the root diameter. \( L \) is defined by:

\[
L = \frac{a_{\text{fin}}}{d_o}
\]

(3.83)

where \( a_{\text{fin}} \) is the area of one side of one fin,

\[
a_{\text{fin}} = \frac{\pi}{4}(d_o^2 - d_r^2)
\]

(3.84)

The fin efficiency can be calculated from:

\[
\Phi = \frac{1}{1 + \frac{m^2}{3} \sqrt{\frac{d_o}{d_r}}}
\]

(3.85)

where

\[
m = H \sqrt{\frac{2}{\left( \frac{1}{R_{fo}} + R_{fo} \right) k_{in} Y}}
\]

(3.86)

also,

\[
A_{\text{fin}} = \frac{\pi}{2}(d_o^2 - d_r^2)N_f L
\]

(3.87)

where \( N_f \) is the number of fins per unit length and \( L \) is the finned length of the tube,

\[
A_{eq} = \Phi A_{\text{fin}} + \pi d_r L \left( \frac{s}{s + Y} \right)
\]

(3.88)

and

\[
A_{\text{root}} = \pi d_r L \left( \frac{s}{s + Y} \right)
\]

(3.89)

The above set of equations is quite accurate for calculating the heat transfer coefficient on Trufin. However, it is also rather awkward to use, and it has been found as a practical matter that a much simpler and nearly as good a result can be obtained by the use of Eq. (3.79) using \( d_r \) as the diameter and applying the value of \( h_c \) thus found to the entire outside finned tube surface \( A_o \).
Alternatively, Eq. (3.80) can be used to calculate $h_0$. Use of either equation is in principle a trial and error calculation, and the actual procedure is later illustrated by an example problem.

In the application of all of these equations, it must be remembered that Trufin tubes should not be used for condensing steam. Water has a high surface tension and the condensate film will bridge the fins. The thick layer of liquid thus held in place on the tube acts as an insulator; the resulting heat transfer rate per unit length of tube is substantially less than for a plain tube. It is not clear at what value of the surface tension this effect becomes significant. Condensates with surface tensions as high as 25-30 dyne/cm have been condensed on finned tubes with no reported difficulty; this includes the usual design range for hydrocarbons, alcohols, refrigerants, etc., with surface tensions from 10 to 25 dyne/cm. However, there is a severe penalty for the condensation of water at atmospheric pressure, at which the surface tension is about 60 dyne/cm. No data or reported experience appears to exist in between.

3.3.7. Filmwise Condensation on Tube Banks

1. Effect of number of rows of tubes.

In any practical condenser design the condensate formed on tubes near the top of the bundle will fall on tubes lower in the bank and presumably modifying the heat transfer coefficient. Nusselt (7) assumed undisturbed laminar flow on each successive tube and found theoretically that the average condensing heat transfer coefficient for a bank of horizontal plain tubes is given by:

$$h_{c,N} = h_{c,1}N^{-1/4}$$

(3.91)

where $h_{c,1}$ is the heat transfer coefficient for one row calculated by the previous equations and $N$ is the number of tubes in one vertical row. For a reasonable size process condenser $N$ may be 20 or 30, and this equation predicts a very severe penalty compared to a single tube. Kern (21) recommended an exponent of (-1/6), again for plain tubes. However, Short and Brown (22) experimentally found no net penalty against the single tube coefficient in a single row 20 tubes high.

The latter result is generally borne out in process experience, and current design practice is to assume that the average coefficient for the entire tube bank is the same as for a single tube. The explanation generally advanced is that rippling, splashing, and turbulence induced by liquid falling from one tube to the next overcomes the possible disadvantages of an increasing liquid loading on each tube.

2. Effect of vapor shear.

It is well known that vapor flowing at a high velocity across the tubes can increase the heat transfer coefficient significantly above the theoretical gravity-driven flow model of Nusselt. However, very few data actually exist to estimate the significance of the effect and most of these data are proprietary. One
widely-cited graph, Fig. 3.24, indicates that the heat transfer coefficient becomes greater than the Nusselt value at vapor Reynolds numbers above about 30,000, increasing to about 10 times as much at $Re_v = 100,000$. However, all data are for plain tube banks and often at vapor velocities so high that excessive pressure drops would be required to achieve significant improvement. Until more data are available, it appears prudent to calculate condensing coefficients as if the flows were purely gravity-driven.

### 3.3.8. Pressure Drop during Shell Side Condensation

Very few data have been published for pressure drop during condensation on the shell-side of the shell and tube heat exchanger. Diehl and Unruh (23, 24) presented a correlation for two-phase adiabatic (i.e., no phase change) pressure drop across several tube banks in both vertical downwards and horizontal cross flow for an ideal tube bank.

Making certain assumptions, Brooks (25) worked from the Diehl-Unruh correlations to obtain a correction factor converting the calculated shell-side pressure drop for the all-vapor flow to the pressure drop for a condenser with a saturated vapor inlet and an exit vapor quality $x_o$. This figure is given here as Fig. 3.25.

In order to use Fig. 3.25, one must first calculate the shell side pressure drop for the all-vapor flow, using the method given in Chapter 2. Since that method is quite lengthy to present, it will not be repeated here. Then the pressure drop for the all-vapor flow is multiplied by the correction factor $\Phi_{gtt}$, from Fig. 3.25 for the appropriate exit quality from the condenser.

Several assumptions have been made in obtaining Fig. 3.25, among the more important being:

1. The shell-side vapor flow is always turbulent.

2. All tube bank layouts have the same correction factor. (Diehl and Unruh found that the 45° banks give somewhat higher pressure drops than the 60° and 90° banks in the liquid-rich end, where the pressure gradient is less anyway. The present curve is based on the mean of the results.)

3. The effect of two-phase flow on pressure drop in the windows is similar to that in crossflow.

4. The rate of condensation is constant through the length of the bundle. A partial condenser that has a higher condensation rate near the entrance should give a lower pressure drop than estimated by Fig. 3.25 and vice versa.

It should be recognized that the calculation of shell-side pressure drop is at best a rough estimation and that too little testing has been done against actual condensers to allow for a more precise method.
3.4. Examples of Design Problems for Low- and Medium-Finned Trufin in Shell and Tube Condensers

3.4.1. Condenser Design for a Pure Component: Example Problem

Statement of Problem

Design a condenser to condense 2,630,000 lb/hr of propane at 148°F and 200 psia (saturation temperature is 105.2°F) using water at 86°F. Maximum allowable pressure drop is 10 psi for the propane and 25 psi for the water. Minimum water velocity is 8 ft/sec. Fouling resistances are 0.001 hrft²°F/Btu for the water and 0.0003 for the propane, each based upon the respective heat transfer areas. An E or J shell design is to be used, with maximum allowable baffle spacing and cut. Wolverine S/T Trufin tubes, 3/4 in. by 16 BWG, 19 fins/in., of 70/30 CuNi are to be used. Maximum allowable tube length is 80 feet. Fixed tube sheet construction is satisfactory.

Some Comments Upon the Design

This problem is similar to an actual design performed for the propane condensers for a large LNG plant using brackish water as a coolant. The high water velocity is intended to minimize fouling, and 70/30 CuNi is selected for its resistance to corrosion. The original design was for a rod baffle exchanger, but a conventionally baffled shell is chosen here solely to illustrate the shell side pressure drop calculation.

Preliminary Estimate of Size

1. Heat duty

The heat duty is composed of two parts: the sensible heat of desuperheating the vapor from 148°F to 105.2°F and the latent heat of condensation at 105.2°F.

Desuperheating:

\[ Q_{DSH} = W C_p (T_{SH} - T_{SAT}) \]  \hspace{1cm} (3.92)

\[ Q_{DSH} = (2,630,000 \text{ lb/hr})(0.39 \text{ Btu/lb}^\circ\text{F})(148 - 105.2)^\circ\text{F} = 4.39 \times 10^7 \text{ Btu/hr} \]
Laten heat:

\[ Q_c = (2,630,000 \text{ lb/hr})(138.1 \text{ Btu/lb}) = 3.63 \times 10^8 \text{ Btu/hr} \]  

(3.62)

Total duty = \( 4.07 \times 10^8 \text{ Btu/hr} \)

It will be assumed that the total duty is transferred by condensation at a constant temperature of 105.2°F

2. Mean temperature difference

A design outlet water temperature must be chosen. Try 95°F for a first set of calculations.

Then

\[
LMTD = \frac{95 - 86}{\ln \left( \frac{105.2 - 86}{105.2 - 95} \right)} = 14.2^\circ F
\]

(3.56)

Since the condensing side is isothermal, \( F = 1.00 \).

We may also calculate the water requirement at this time:

\[
w_{H_2O} = \frac{4.07 \times 10^8 \text{ Btu/hr}}{(9.0^\circ F)(1.00 \text{ Btu/lb} \cdot \text{°F})} = 4.52 \times 10^7 \text{ lb/hr}
\]

(3.64)

3. Overall heat transfer coefficient

Estimate the various coefficients and resistances, based on their respective heat transfer areas:

Condensing: \( h_o = 500 \text{ Btu/hr ft}^2 \text{°F}, \) based on total outside heat transfer area.

Fouling, outside: \( R_{fo} = 3.0 \times 10^{-4} \text{ hr ft}^2 \text{°F/Btu}, \) same basis

Fin resistance: \( R_{fin} = 7.1 \times 10^{-4} \text{ hr ft}^2 \text{°F/Btu}, \) same basis

0.065 in.

Wall resistance:

\[
R_w = \frac{0.065 \text{ in.}}{(12 \text{ in./ft})(17 \text{ Btu/hr ft}^2 \text{°F})}
\]

= 0.00032 \text{ hr ft}^2 \text{°F/Btu}, based on the mean wall diameter,

Water: \( h_i = 1600 \text{ Btu/hr ft}^2 \text{°F}, \) based on the inside tube area from Fig. 2.19

Fouling, inside: \( R_i = 0.001 \text{ hr ft}^2 \text{°F/Btu}, \) same basis

Combining these, with the appropriate corrections to put the overall coefficient on the basis of the total outside heat transfer area gives:
\[ U_o = \frac{1}{500 + 0.0003 + 7.1 \times 10^{-4} + 0.00032} = 96.4 \text{ Btu/hr ft}^2\text{°F} \]

\[ = \frac{1}{1 + 0.001} \left( \frac{0.503}{0.573} \right) + \left( \frac{1}{1600} + 0.001 \right) \left( \frac{0.503}{0.130} \right) \]

(3.55)

based on total outside heat transfer area.

4. Calculation of area

\[ A_o = \frac{4.07 \times 10^8 \text{ Btu/hr}}{(96.4 \text{ Btu/hr ft}^2\text{°F})(14.2 \text{°F})} = 2.97 \times 10^5 \text{ ft}^2 \]

(3.54)

5. Estimation of shell dimensions

We may use the methods described in Chapter 2. Specifically, we may use Fig. 2.26, but first we must compute the effective area \( A'_o \) with which to enter the chart:

\[ A'_o = A_o F_1 F_2 F_3 F_4 \]

where \( A_o \) is the actual area required in the heat exchanger, \( 2.97 \times 10^5 \text{ ft}^2 \) for the present case

\( F_1 \) is a correction factor for the unit cell array. Since there is no apparent reason why a 3/4 in. OD tube on 15/16 in. triangular layout will not serve, \( F_1 = 1.00 \).

\( F_2 \) is a correction factor for the number of tube passes. If possible, we will use one pass, so \( F_2 = 1.00 \) for present.

\( F_3 \) is a correction factor for the shell construction. Fixed tube sheet construction is satisfactory, so \( F_3 = 1.00 \).

\( F_4 \) is the correction factor for the specific fin geometry and density. For 3/4 in. O.D. 19 fins/in. construction, \( F_4 = 1.00 \).

so

\[ A'_o = 2.97 \times 10^5 \text{ ft}^2 \text{ (1.00)(1.00)(1.00)(1.00)} = 2.97 \times 10^5 \text{ ft}^2 \]

Referring to Figure 2.26, we see that a combination of diameter and length (on the figure) that will meet this requirement is a 120 in. I.D. shell, with tubes about 41 feet long. Referring to Table 2.6, we see that the tube count for a one pass exchanger is about 14,500. This gives a total area of

\[ A_o = (14,500 \text{ tubes})(0.503 \text{ ft}^2/\text{ft})(41 \text{ ft}) = 299,034 \text{ ft}^2 \], compared to the 297,000 \text{ ft}^2 estimated. This would allow a 40.33 ft long tube.

Check the water side velocity:
\[ V_{H,O} = \frac{\left(4.52 \times 10^7 \text{ lb/hr} \left(144 \text{ in}^2 / \text{ft}^2\right)\right)}{14,500 \left(62.1 \text{ lb/ft}^3 \left(3600 \text{ sec/hr} \left(0.195 \text{ in.}^2\right)\right)\right)} = 10.3 \text{ ft/sec} \]

This velocity is above the stated minimum; check the pressure drop:

\[ \text{Re}_i = \frac{(0.508/12) \left(62.1 \text{ lb/ft}^3 \left(10.3 \text{ ft/sec} \left(3600 \text{ sec/hr}\right)\right)\right)}{1.79 \text{ lb/ft/hr}} = 54,450 \quad (2.19) \]

\[ f_i = 0.0046 \text{ from Fig. 2.20} \]

\[ \Delta p_f = \frac{2(0.0046) \left(62.1 \text{ lb/ft}^3 \left(10.4 \text{ ft/sec} \left(40.33 \text{ ft}\right)\right)\right)}{\left(0.508/12\right) \left(32.2 \frac{\text{lb}_{w} \text{ ft}}{\text{lb}_{f} \text{ sec}^2} \left(144 \text{ in}^2 / \text{ft}^2\right)\right)} = 12.45 \text{ psi} \quad (2.26) \]

(neglecting the Sieder-Tate term)

The entrance pressure loss is:

\[ \Delta p_{ent} = \frac{3 \left(62.1 \text{ lb/ft}^3 \left(10.3 \text{ ft/sec}\right)^2\right)}{2 \left(32.2 \frac{\text{lb}_{w} \text{ ft}}{\text{lb}_{f} \text{ sec}^2} \left(144 \text{ in}^2 / \text{ft}^2\right)\right)} = 2.1 \text{ psi} \quad (2.25) \]

so the calculated tube-side pressure drop is within the design limit. Proceed with this design.

Shells of this diameter fall outside standard TEMA specifications on baffle spacing. It should be mechanically conservative to design to the maximum unsupported span of 52 in. (i.e., baffle spacing of 26 in. for a fully tubed bundle) recommended in Table R-4.52 of Ref. (1). If this gives excessive velocities or pressure drops, we may then switch the design to a J or an X shell, or consider a rod baffle design.

**Heat Transfer Calculation**

Having a general idea of the design, it is now necessary to check the specific values of the heat transfer coefficients.

1. The condensing coefficient

First, calculate the coefficient by Eq. (3.80). \( W \) is the mass of vapor condensed per tube in unit time:

\[ W = \frac{2,630,000 \text{ lb/hr}}{14,500 \text{ tubes}} = 181.3 \text{ (lb/hr)/tube} \]

Then
We may check this against the Beatty-Katz prediction, Eq. 3.81 to 3.89). The various quantities are:

\[ A_{\text{root}} = \pi \left( \frac{0.638}{12} \text{ ft} \right) \left( 40.33 \text{ ft} \right) = 6.7 \text{ ft}^2 \]

\[ A_{\text{fin}} = \pi \left[ \frac{0.750^2 - 0.638^2}{144} \right]^{1/2} \left( 19 \right) \left( 12 \text{ fins/ft} \right) \left( 40.33 \text{ ft} \right) = 15.6 \text{ ft}^2 \]

\[ m = \left( \frac{0.056}{12} \text{ ft} \right) \left[ \frac{2}{\frac{1}{511} + 0.0003} \right] \left( \frac{\text{Btu/hr}^2\text{ F}}{17 \text{ Btu/hr}^2\text{ F}} \right) \left( \frac{0.011}{12 \text{ ft}} \right) = 1.11 \]

In principle, this quantity must be recalculated using the new value of \( h_c \) until the solution converges.

\[ \Phi = \frac{1}{1 + \frac{(1.11)^2}{2}} \left( \frac{0.750 \text{ in.}}{0.638 \text{ in.}} \right) = 0.692 \]

\[ A_{\text{eq}} = 0.692 \left( 15.6 \text{ ft}^2 \right) + \pi \left( \frac{0.638}{12} \text{ ft} \right) \left( 40.33 \text{ ft} \right) \left( \frac{0.0265 \text{ in.}}{0.056 \text{ in.}} \right) = 14.0 \text{ ft}^2 \]

\[ a_{\text{fin}} = \pi \left( \frac{0.750^2 - 0.638^2}{144} \right) \text{ ft}^2 = 8.5 \times 10^{-4} \text{ ft}^2 \]

\[ L = \frac{8.5 \times 10^{-4} \text{ ft}^2}{\left( \frac{0.750}{12} \text{ ft} \right)} = 0.0136 \text{ ft} \]

\[ \left( \frac{1}{d_{\text{eq}}} \right)^{1/4} = 1.3 \left( 0.692 \right) \left( \frac{15.6 \text{ ft}^2}{14.0 \text{ ft}^2} \right)^{1/4} + \frac{1}{0.0136} \left( \frac{12 \text{ ft}}{14.0 \text{ ft}^2} \right)^{1/4} \left( \frac{0.638}{0.315} \right)^{1/4} = 3.93 \text{ ft}^{-1/4} \]

Estimate wall temperature to be 100°F
This value should be converged as indicated above and will be somewhat smaller than indicated. Even so, it will be substantially higher than the 511 Btu/hr ft\(^2\circ\)F obtained from the unmodified Nusselt equation. However, the effect on the overall coefficient will be quite small (about 5 per cent increase) because the condensing process is only a small part (about 15\%) of the total resistance. In view of the other uncertainties in the calculations (and realizing that the water side fouling resistance is both the most uncertain term and the largest single resistance), this difference is probably not enough to worry about.

Now compute the overall heat transfer coefficient based upon the revised values:

Condensing: \(h_c = 510\) Btu/hr ft\(^2\circ\)F, based on total outside heat transfer area, \[\text{Fouling, outside: } R_{fo} = 0.0003\text{ hr ft}\(^2\circ\)F/Btu, same basis,}\]

Fin resistance: \(R_{fin} = 7.1 \times 10^{-4}\) hr ft\(^2\circ\)F/Btu, same basis,

Wall resistance: \(R_w = 0.00032\) hr ft\(^2\circ\)F/Btu, based on mean wall diameter,

Water: \(h_i = 1990\) Btu/hr ft\(^2\circ\)F, based on inside area from Fig. 2.19,

Fouling, inside: \(R_{fi} = 0.001\) hr ft\(^2\circ\)F/Btu, same basis

Then,

\[
U_o = \frac{1}{\frac{1}{510} + 0.0003 + 7.1 \times 10^{-4} + 0.00032 \left[ \frac{0.503}{\pi \left( \frac{0.573}{12} \right)} \right] + \left( \frac{1}{1990} + 0.001 \right) \left( \frac{0.503}{0.136} \right)} = 101.4\text{ Btu/hr ft}\(^2\circ\)F}
\]

And the corresponding area is

\[
A_o = \frac{4.07 \times 10^8\text{ Btu/hr}}{(101.4\text{ Btu/hr ft}\(^2\circ\)F)(14.2\circ F)} = 282,700\text{ ft}^2
\]

\[
L = \frac{282,000\text{ ft}^2}{(14,500\text{ tubes})(0.503\text{ ft}^2/\text{ft})} = 38.8\text{ ft}
\]

Say 39 ft. for further calculations.
Shell-Side Pressure Drop Calculation

So far we have established that the heat transfer characteristics and the tube-side pressure drop and velocity are satisfactory. We still need to calculate the shell-side pressure drop. The basic procedure is to calculate the shell-side pressure drop as if the vapor phase flowed uncondensed for the entire length (using the Delaware method as given in Chapter 2 and then to correct it by a two-phase multiplier as shown in Fig. 3.25.

1. Basic geometrical data
   
   Tube outside diameter: \( d_o = 0.750 \text{ in.} \)
   
   Tube root diameter: \( d_r = 0.638 \text{ in.} \)
   
   Fin spacing: \( s = 0.0265 \text{ in.} \)
   
   Fin thickness: \( Y = 0.011 \text{ in.} \)
   
   Tube layout: Equilateral triangular with 1.5/16 in. = 0.9375 in. pitch
   
   Shell inside diameter: \( D_i = 120 \text{ in.} \)
   
   Shell outer tube limit: \( D_{otl} = 119 \text{ in.} \)
   
   Effective tube length: \( L = 39 \text{ ft.} \)
   
   Baffle cut: \( \ell_c = 58 \text{ in.} \)
   
   Baffle spacing: \( \ell_x = 26 \text{ in.} \)

   Number of sealing strips per side: \( N_{ss} = 0 \)

2. Shell-side geometrical parameters
   
   a. Total number of tubes in the exchanger: \( N_t = 14,500 \)
   b. Tube pitch parallel to flow: \( p_p = 0.814 \text{ in.} \)
      Tube pitch normal to flow: \( p_n = 0.469 \text{ in.} \)
   c. Number of tube rows crossed in one crossflow section:
      \[
      N_c = \left( \frac{120 \text{ in.}}{0.814 \text{ in.}} \right) - 2 \left( \frac{58 \text{ in.}}{120 \text{ in.}} \right) = 5
      \]
      (2.38)
   d. Fraction of total tubes in crossflow:
      \[
      \frac{\ell_c}{D_i} = \frac{58 \text{ in.}}{120 \text{ in.}} = 0.48
      \]
From Fig. 2.28

\[ F_c = 0.05 \]

e. Number of effective crossflow rows in each window:

\[ N_{cw} = \frac{0.8(58 \text{ in.})}{0.814 \text{ in.}} = 57 \quad (2.40) \]

f. Number of baffles:

\[ N_b = \frac{39}{(26 \text{ in.})} - 1 = 18 \quad (2.41) \]

An even number also means that the nozzles are on opposite sides of the shell as required in this case.

g. Crossflow area at centerline:

\[
S_m = (26 \text{ in.}) \times \left\{120 \text{ in.} - 119 \text{ in.} + \frac{119 \text{ in.} - 0.750 \text{ in.}}{0.9375 \text{ in.}} \left[ (0.9375 \text{ in.} - 0.750 \text{ in.}) + 2(0.056 \text{ in.}) \left( \frac{0.0265 \text{ in.}}{0.0265 \text{ in.} + 0.011 \text{ in.}} \right) \right]\right\}
\]

\[ S_m = 900 \text{ in.}^2 \]

h. Fraction of crossflow area available for bypass flow:

\[ F_{sbp} = \frac{(120 \text{ in.} - 119 \text{ in.})(26 \text{ in.})}{(900 \text{ in.}^2)} = 0.029 \quad (2.44) \]

i. Tube-to-baffle leakage area for one baffle:

\[ S_{tb} = 0.0184(14,500)(1 + 0.05) = 280 \text{ in.}^2 \quad (2.46) \]

j. Shell-to-baffle leakage area for one baffle: Assume diametral shell-to-baffle clearance = 0.700 in.

\[ S_{sb} = \frac{(120 \text{ in.})(0.700 \text{ in.})}{2} \left[ \pi - \cos^{-1} \left( 1 - \frac{2(58 \text{ in.})}{120 \text{ in.}} \right) \right] = 67.4 \text{ in.}^2 \]

k. Area for flow through window:
\[ S_{wg} = \frac{(120 \text{ in.})^2}{4} \left[ \cos^{-1} \left( \frac{1 - 2 \left( \frac{58 \text{ in.}}{120 \text{ in.}} \right)}{1 - 2 \left( \frac{58 \text{ in.}}{120 \text{ in.}} \right)} \right] \left[ 1 - 2 \left( \frac{58 \text{ in.}}{120 \text{ in.}} \right) \right] \sin \left[ \cos^{-1} \left( \frac{1 - 2 \left( \frac{58 \text{ in.}}{120 \text{ in.}} \right)}{1 - 2 \left( \frac{58 \text{ in.}}{120 \text{ in.}} \right)} \right] \right] \] (2.50)

\[ = 5,415 \text{ in.}^2 \]

\[ S_{wt} = \frac{14,500}{8} (1 - 0.05) \pi (0.750 \text{ in.})^2 = 3,043 \text{ in.}^2 \] (2.51)

\[ S_w = 5,415 \text{ in.}^2 - 3,043 \text{ in.}^2 = 2,372 \text{ in.}^2 \] (2.49)

3. Shell-side pressure drop calculation

a. Calculate shell-side Reynolds number:

\[ \text{Re}_s = \left( \frac{0.0638}{12} \right) \left( \frac{2,630,000 \text{ lb}}{\text{hr}^2} \right) \left( 0.021 \frac{\text{lb}}{\text{hr}^2} \right) \left( \frac{900}{144} \text{ ft}^2 \right) = 1.07 \times 10^6 \] (2.54)

It is necessary to guess a friction factor for this case as being about 0.2. (See Fig. 2.17)

b. Pressure drop for an ideal crossflow section:

\[ \Delta p_{b,l} = \frac{4(0.2)(2,630,000)^2}{2(1.85) \left( 4.17 \times 10^8 \frac{\text{lb}}{\text{hr}^2} \right) \left( 900 \frac{144}{144} \text{ ft}^2 \right)} = 459 \text{ lb} / \text{ft}^2 = 3.2 \text{ psia} \] (2.57)

The viscosity gradient correction is ignored in condensing flows.

c. Pressure drop for an ideal window section:

\[ \Delta p_{w,d} = \frac{(2,630,000 \text{ lb} / \text{hr})^2}{2 \left( 4.17 \times 10^8 \frac{\text{lb}}{\text{hr}^2} \right) \left( 900 \frac{144}{144} \text{ ft}^2 \right) \left( 2,372 \frac{144}{144} \text{ ft}^2 \right)} = 1,576 \text{ lb} / \text{ft}^2 = 10.9 \text{ psia} \] (2.58)

d. Correction factor for baffle leakage:

\[ \frac{S_{sb} + S_{th}}{S_m} = \frac{67.4 \text{ in.}^2 + 280 \text{ in.}^2}{900 \text{ in.}^2} = 0.386 \]

\[ \frac{S_{sb}}{S_{sb} + S_{th}} = \frac{67.4 \text{ in.}^2}{67.4 \text{ in.}^2 + 280 \text{ in.}^2} = 0.19 \]

From Fig. 2.38
\[ R_e = 0.48 \]

e. Correction factor for bundle bypass:

\[ F_{sbp} = 0.029 \]

\[ N_{ss} = 0 \]

From Fig. 2.39

\[ R_p = 0.90 \]

f. Total shell-side pressure drop for an all vapor flow:

\[ \Delta p_s = \left[ (18 - 1)(3.2 \text{ psi})(0.90) + 18(10.9 \text{ psi})(0.48) + 2(3.2 \text{ psi})(0.90)(1 + \frac{57}{2}) \right] = 215 \text{ psi} \quad (2.60) \]

for an all-vapor flow.

g. Correction for a condensing flow:

From Fig. 3.25, we find that a totally-condensed flow has a pressure drop 0.29 times the pressure drop for the corresponding all vapor-flow, so the estimated pressure drop for the proposed condenser design would be 62.2 psi.

This value is substantially greater than the allowable pressure drop, and some means must be found to reduce this value.

**Alternative Designs for This Case**

Besides the excessive pressure drop calculated for the conventional E shell design in this case, there are several other concerns. Considering the high velocities, tube vibration is a definite possibility, especially near the inlet, and an annular distributor (vapor belt) should be specified for all E and J shells (including rod baffle designs). For an X shell (which may also be a rod baffle), a single oversized ("bathtub") nozzle should be used running nearly the full length of the top of the shell, or multiple conventional nozzles may be used along the top at some increase in the piping cost.

To deal with shell-side pressure drop itself, a conventionally baffled J shell design (with otherwise the same geometry as the computed example) would probably suffice. A J shell will reduce the shell-side pressure drop by a factor of 6 to 8 compared to an identical E shell. The decrease in shell-side velocity will not affect the calculated (design) value of the condensing coefficient, because no credit is taken for vapor shear enhancement.

Alternatively, an E shell using rod baffle construction would also reduce the pressure drop to within acceptable limits. This pressure drop could be estimated by calculating the longitudinal flow pressure drop through the tube array using an equivalent diameter based upon the outside diameter of the tubes; the same \( \Delta p \) correction factor for condensing flow across tubes, Fig. 3.25 could be used with reasonable accuracy.

Finally, an X shell using same shell diameter and length could be used either with full circle tube supports or rod baffle supports, as long as the vapor was well distributed along the length of the bundle as noted above.
3.4.2. Condenser Design for a Multi-component Mixture: Example Problem

Special Considerations of Multi-component Condensation

The mechanisms in condensing a multi-component mixture were described qualitatively in Chapter 1. The important differences compared to pure component condensation may be summarized as follows:

1. The heavier components preferentially condense first so that the compositions of each phase are changing from point to point.

2. As condensation proceeds, the condensing temperature decreases and the remaining vapor must be sensibly cooled also. The corresponding sensible heat transfer duty must be transferred from the vapor by a sensible heat transfer coefficient which is generally quite low compared to the condensing coefficient.

The multi-component condensation process has been analyzed in fundamental terms by Krishna and Panchal (26), but the use of these methods in design has not yet been completed and in any case requires a computer. A more heuristic procedure was proposed by Silver (27) and put in suitable form for condensers with multiple coolant passes by Bell and Ghaly (28). Variations of this method are in use in various computer-based condenser rating procedures, but even the simplified method is extremely tedious for hand calculations. So in the following section a reduced form of the Silver method is illustrated; this method will be generally suitable for well-behaved cases - ones in which the condensing curve (of temperature vs. heat release) is nearly linear or slightly concave upwards, in which the properties do not change greatly from start to finish, and in which the coolant temperature does not approach the vapor temperature too closely.

Under these conditions, the design integral from (27) can be replaced by the following approximation:

\[ A_o = \left[ \frac{1}{{U_o'}} + \frac{Z}{h_{sv}} \right] \frac{Q_T}{MTD} \]  

(3.93)

The individual terms in Eq. 3.93 are defined as follows:

\[ U_o' = \frac{A_o}{h_i A_i} + \frac{1}{R_{fi}} + \frac{A_o}{k_w A_w} + \frac{\Delta x_w}{A_o} + \frac{R_{fin} + R_{fo} + 1}{h_o} \]  

(3.94)

The condensing heat transfer coefficient, \( h_o \), is calculated from the methods of this section, such as Eq. (3.80).

\[ Z = \frac{Q_{sv}}{Q_T} \]  

(3.95)

where \( Q_{sv} \) is the sensible heat duty removed by cooling the vapor and can be calculated approximately by the following equation:

\[ Q_{sv} = \bar{C}_{pv} \left[ \frac{W_{v_{out}}}{h_{v_{out}}} + \frac{1}{2} \left( W_{v_{in}} - W_{v_{out}} \right) \right] \left( T_{v_{in}} - T_{v_{out}} \right) \]  

(3.96)
and $Q_T$ is the total heat transferred in the condenser. Then, $h_{sv}$ is the sensible heat transfer coefficient calculated at the half-condensed point and as if only the vapor phase were flowing. Finally, the MTD is calculated from the LMTD and the configuration correction factor (for multiple coolant passes) using the vapor and coolant terminal temperatures.

The application of these equations is illustrated in the following example problem.

**Statement of the Problem**

A saturated vapor mixture composed of 0.6 mol fraction nC$_5$ and 0.4 nC$_4$ at a pressure of 50 psia is to be totally condensed at the rate of 120,000 lb/hr on the shell-side of a horizontal E-shell and tube heat exchanger using Wolverine S/T low-finned Trufin with cooling water available at 85 F. Fixed tube sheet and low carbon steel construction will be acceptable. Pressure drop for the vapor side should not exceed 5 psi and for the water 10 psi.

Fouling resistances are 0.001 hr*ft$^2$/°F/Btu for the water and 0.0005 for the hydrocarbon mixture. Wolverine S/T Trufin, 3/4 in. x 16 BWG, 19 fins/in. of low carbon steel are to be used.

**Physical properties:**

The physical properties for the pure components are taken from Perry (29) and the properties of the mixture are calculated using the mixing rules in the same source. These values will be assumed constant throughout the problem:

<table>
<thead>
<tr>
<th></th>
<th>Liquid</th>
<th>Vapor</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density, lb/ft$^3$</td>
<td>33.4</td>
<td>0.57</td>
</tr>
<tr>
<td>Viscosity, lb/hr ft</td>
<td>0.375</td>
<td>0.017</td>
</tr>
<tr>
<td>Specific heat, Btu/lb°F</td>
<td>0.58</td>
<td>0.42</td>
</tr>
<tr>
<td>Thermal conductivity, Btu/hr ft°F</td>
<td>0.077</td>
<td>0.0098</td>
</tr>
</tbody>
</table>

The condensing curve can be constructed from vapor pressure and enthalpy data from the same source. This curve is shown in Fig. 3.26.

Water properties are given at 100°F and assumed constant throughout.

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Density, lb/ft$^3$</td>
<td>62.0</td>
</tr>
<tr>
<td>Viscosity, lb/hr ft</td>
<td>1.63</td>
</tr>
<tr>
<td>Specific heat, Btu/lb°F</td>
<td>1.00</td>
</tr>
<tr>
<td>Thermal conductivity, Btu/hr ft°F</td>
<td>0.361</td>
</tr>
</tbody>
</table>
### Tube Characteristics

<table>
<thead>
<tr>
<th>Characteristic</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outside diameter</td>
<td>0.750 in.</td>
</tr>
<tr>
<td>Inside diameter</td>
<td>0.495 in.</td>
</tr>
<tr>
<td>Root diameter</td>
<td>0.625 in.</td>
</tr>
<tr>
<td>Fin height</td>
<td>0.052 in.</td>
</tr>
<tr>
<td>Fin thickness</td>
<td>0.011 in.</td>
</tr>
<tr>
<td>Outside heat transfer area</td>
<td>0.503 ft²/ft</td>
</tr>
<tr>
<td>Inside heat transfer area</td>
<td>0.1303 ft²/ft</td>
</tr>
<tr>
<td>Outside/inside area ratio</td>
<td>3.86</td>
</tr>
<tr>
<td>Inside flow area per tube</td>
<td>0.195 in.²</td>
</tr>
<tr>
<td>Thermal conductivity</td>
<td>26 Btu/hr ft°F</td>
</tr>
<tr>
<td>Fin resistance</td>
<td>3.1x10⁻⁴ hr ft²°F/Btu</td>
</tr>
</tbody>
</table>

*Fig. 3.26 Condensing curve for C₄ - C₅ example problem.*
Preliminary Estimate of Design

1. Heat duty

\[ Q = 1.798 \times 10^7 \text{ Btu/hr (from Fig. 3.26)} \]

2. Mean temperature difference

Try a design outlet water temperature of 115°F.

\[ LMTD = \frac{(152 - 115) - (136 - 85)}{\ln\left(\frac{152 - 115}{136 - 85}\right)} = 43.6°F \]

Assume multiple tube passes

\[ P = \frac{115 - 85}{152 - 85} = 0.448 \]
\[ R = \frac{152 - 136}{115 - 85} = 0.533 \]
\[ F = 0.955 \]
\[ MTD = 0.955 \times (43.6) = 41.6°F \]

3. Estimated overall heat transfer coefficient, based on outside surface (not to be confused with \( U' \)):

\[
U_o = \frac{1}{\frac{1}{\frac{1}{h_o} + R_{fo} + R_{fin} + \frac{\Delta x_w}{k_w} A_o + \left(\frac{1}{h_i} + R_{fi}\right) \frac{A_o}{A_i}}} = \frac{1}{1 + \frac{5 \times 10^{-4} + 3.1 \times 10^{-4} + 0.065}{250} + \frac{0.001}{0.503} + \frac{0.503}{0.1303}} = 75.5 \text{ Btu/hr ft}^2\cdot\text{°F}
\]

Note that in order to reflect the extra resistance of the vapor phase, the condensing coefficient was estimated at 250 Btu/hr ft²°C, rather than the 400 typical of pure component condensation under these conditions.

4. Calculation of area

\[ A_o = \frac{1.789 \times 10^7}{41.6(75.5)} = 5696 \text{ ft}^2, \text{ estimated total outside finned surface.} \]

5. Estimation of shell dimensions. Again, we may use Fig. 2.26 of chapter 2.
\[ A'_o = A_o F_1 F_2 F_3 F_4 \]

\[ A_o = 5696 \text{ ft}^2 \]

\[ F_1 = 1.00 \text{ (3/4 in. tube on 15/16 in. triangular pitch)} \]

\[ F_2 = 1.03 \text{ (Assume - for the moment - 2 tube passes in a 31 in. shell)} \]

\[ F_3 = 1.00 \text{ (Fixed tube sheet)} \]

\[ F_4 = 1.00 \text{ (S/T Trufin 19 fins/in.)} \]

\[ A_o' = 5696 \times 1.00 \times 1.03 \times 1.00 \times 1.00 \]

\[ A_o' = 5867 \text{ ft}^2 \]

The possibilities (from Fig. 2.26) are:

<table>
<thead>
<tr>
<th>Shell ID, in.</th>
<th>Effective Tube Length, ft.</th>
</tr>
</thead>
<tbody>
<tr>
<td>35</td>
<td>10</td>
</tr>
<tr>
<td>33</td>
<td>11</td>
</tr>
<tr>
<td>31</td>
<td>13</td>
</tr>
<tr>
<td>29</td>
<td>15</td>
</tr>
<tr>
<td>27</td>
<td>17.5</td>
</tr>
<tr>
<td>25</td>
<td>20</td>
</tr>
<tr>
<td>23 1/4</td>
<td>23.5</td>
</tr>
</tbody>
</table>

Choose the 31 in. ID case for further analysis.

Check water-side velocity: 2 passes:

\[ w_{H_2O} = \frac{1.798 \times 10^7 \text{ Btu/hr}}{(1.00 \text{ Btu/lb } 30^\circ F)} = 599,000 \text{ lb/hr} \]

For two tube passes, \( N_t = 878 \), or 439/pass

So:

\[ V_{H_2O} = \frac{(599,000 \text{ lb/hr})(144 \text{ in.}^2 / \text{ft}^2)}{439(0.195 \text{ in.}^2)(62.0 \text{ lb/ft}^3)(3600 \text{ sec/hr})} = 4.51 \text{ ft/sec} \]

which is quite acceptable for the moment.

**Thermal Design**

The problem here is to calculate \( h_o \) and \( h_{sv} \) for the shell side and \( h_i \) for the tube-side so that Eq. (3.93) and (3.94) can be evaluated.
1. Calculation of $h_o$:

Using Eq. 3.80 and assuming $L = 13\ ft$.

$$W = \frac{120,000\ lb/\ hr}{878\ tubes} = 136.7\ lb/\ hr\ per\ tube$$

$$h_o = 0.951\ \left(\frac{0.077\ Btu}{hr\ ft^2\ °F}\right)^3\ \left(33.4\ \frac{lb}{ft^3}\right)\ \left(32.8\ \frac{lb}{ft^3}\right)\ \left(4.17 \times 10^8\ \frac{ft}{hr^2}\right)\ (13\ ft)^{1/3}$$

$$= 357\ Btu/\ ft^2\ °F\ ,\ based\ on\ outside\ finned\ tube\ surface\ area.$$

2. Calculation of $h_{sv}$:

The Delaware method will be used based upon the vapor flow rate at the half-condensed point, i.e., 60,000 lb/hr. For a preliminary calculation, a baffle spacing equal to 22.28 in. and a baffle cut of 35 percent of the diameter (10.85 in.) will be assumed. Then the shell-side parameters are: (From Chapter 2)

a. $N_t = 878$

b. $p_o = 0.814\ in.$

c. $p_n = 0.469\ in.$

d. $\ell_c = \frac{10.85\ in.}{31\ in.} = 0.35$

e. $F_c = 40$

f. $N_{cw} = \frac{0.8(10.85\ in.)}{0.814\ in.} = 11$

g. $\frac{13\ ft}{22.80\ in./\ ft} - 1 = 6\ baffles$
g. \[ S_m = (22.28 \text{ in.}) \left\{ 31 \text{ in.} - 30.5 \text{ in.} + \left( \frac{30.5 \text{ in.} - 0.750 \text{ in.}}{0.9375 \text{ in.}} \right) \left( 0.9375 \text{ in.} - 0.750 \text{ in.} \right) + 2(0.052 \text{ in.}) \left( \frac{0.0416 \text{ in.}}{0.0416 \text{ in.} + 0.011 \text{ in.}} \right) \right\} \]
\[ = 202 \text{ in.}^2 \]

h. \[ F_{sbp} = \frac{(31.0 \text{ in.} - 30.5 \text{ in.})22.28}{201.9 \text{ in.}^2} = 0.055 \]

i. \[ S_{ib} = 0.0184 \left( 878 \right) (1 + 0.40) = 22.6 \text{ in.}^2 \]

j. Assume diametral clearance of 0.300 in.
\[ S_{sb} = \frac{(31 \text{ in.})(0.300 \text{ in.})}{2} \left[ \pi - \cos^{-1} \left( 1 - \frac{2(10.85 \text{ in.})}{31 \text{ in.}} \right) \right] = 8.7 \text{ in.}^2 \]

k. \[ S_{wq} = \frac{(31\text{ in.})^2}{4} \left[ \cos^{-1} \left( 1 - \frac{2(10.85 \text{ in.})}{31 \text{ in.}} \right) - \left( 1 - \frac{2(10.85 \text{ in.})}{31 \text{ in.}} \right) \times \sin^{-1} \left( \frac{2(10.85 \text{ in.})}{31 \text{ in.}} \right) \right] \]
\[ = 235.4 \text{ in.}^2 \]

\[ S_{wr} = \frac{878}{8} (1 - 0.40)\pi \left( 0.750 \text{ in.} \right)^2 = 116.4 \text{ in.}^2 \]

\[ S_w = 235.4 \text{ in.}^2 - 116.4 \text{ in.}^2 = 119 \text{ in.}^2 \]

The heat transfer calculation is as follows:

a. \[ Re_s = \frac{0.625 \text{ in.}(60,000 \text{ lb} / \text{ hr})(12 \text{ in.} / \text{ ft})}{(0.017 \text{ lb} / \text{ ft hr})(201.9 \text{ in.}^2)} = 131,000 \]

b. \[ j_s = 0.0039 \]

c. \[ h_{ol} = 0.0039 \left( \frac{0.42 \text{ Btu}/\text{lb}^\circ \text{F}}{201.9 \text{ in.}^2} \right) \left[ \frac{(60,000 \text{ lb} / \text{ hr})(144 \text{ in.}^2 / \text{ ft}^2)}{201.9 \text{ in.}^2} \right] = 0.0098 \text{ Btu} / \text{ hr ft}^\circ \text{F} \]
\[ = 86.6 \text{ Btu/hr ft}^\circ \text{F} \]

d. \[ J_c = 0.85 \]

e. \[ \frac{S_{sb} + S_{ib}}{S_m} = \frac{8.7 \text{ in.}^2 + 22.6 \text{ in.}^2}{201.9 \text{ in.}^2} = 0.155 \]
\[ \frac{S_{sb}}{S_{sb} + S_{ib}} = \frac{8.7 \text{ in.}^2}{8.7 \text{ in.}^2 + 22.6 \text{ in.}^2} = 0.278 \]
\( J_t = 0.775 \)

f. \( \frac{N_{ss}}{N_c} = 0 \); \( F_{abs} = 0.055 \)

\( J_b = 0.93 \)

g. \( h_{sv} = 86.6(0.85)(0.775)(0.93) \text{ Btu/hr ft}^2\text{°F} \)

\[ = 53.1 \text{ Btu/hr ft}^2\text{°F} \text{, based on outside tube surface area.} \]

3. Calculation of \( h_i \): use Fig. 2.19.

\[ h_i = 1.04(1140 \text{ Btu/hr ft}^2\text{°F}) = 1190 \text{ Btu/hr ft}^2\text{°F} \text{, based on inside tube surface area.} \]

4. Calculation of \( U' \): Refer to Equation (3.94).

\[ U' = \left( \frac{1}{1190} + 0.001 \right) \left( \frac{0.503}{0.1303} + \frac{0.065}{26(12)} \right) + 3.1 \times 10^{-4} + 5 \times 10^{-4} + \frac{1}{357} \]

\[ = 86.8 \text{ Btu/hr ft}^2\text{°F} \]

5. Calculation of \( Z \):

\[ Q_{sv} = 0.42 \frac{\text{Btu}}{\text{lb} \cdot \text{°F}} \left[ 0 + \frac{1}{2} \left( 120,000 \frac{\text{lb}}{\text{hr}} - 0 \right) \right] (152°F - 136°F) = 403,200 \text{ Btu/hr} \]

\[ Z = \frac{Q_{sv}}{Q_T} = \frac{403,200 \text{ Btu/hr}}{1.798 \times 10^7 \text{ Btu/hr}} = 0.0224 \]

6. Calculation of \( A_o \): Refer to Eq. (3.93).

\[ A_o = \left( \frac{1}{86.8} + 0.0224 \right) \left( \frac{1.798 \times 10^7}{41.6} \right) = 5162 \text{ ft}^2 \]

compared to the estimated 5696 ft². The actual area provided by the design rated is:

\[ A_o = (878 \text{ tubes})(13 \text{ ft})(0.503 \text{ ft}^2/\text{ft}) = 5741 \text{ ft}^2 \]

which is 10 percent greater than the calculated required area. This is a reasonable safety factor in this kind of problem, and it is suggested that the designer not be tempted to pare the design down.

**Shell-side Pressure Drop Calculations**

The method applied here is to calculate the pressure drop for vapor-only flowing through the shell-side using the Delaware method, followed by application of the Diehl Unruh two-phase correction factor for total condensation.
1. Delaware calculation of $\Delta p_s$, for vapor only.

a. $Re_s = \frac{0.625 \text{ in.}(120,000 \text{ lb/hr})(12 \text{ in./ft})}{(0.017 \text{ lb/ft hr})(201.9 \text{ in.}^2)} = 262,000$

$f = 0.25$

b. $\Delta p_{h,i} = \frac{4(0.25)(120,000 \text{ lb/hr})^2(11)}{2 \left(0.57 \frac{lb}{ft^3}\right) \left(4.17 \times 10^8 \frac{\text{ft lb}}{hr^2 \text{lb}_f} \left(\frac{201.9}{144} \text{ in.}^2 / \text{ft}^2\right)\right)} = 238 \text{ lb}_f / \text{ ft}^2 = 1.65 \text{ psi}$

c. $\Delta p_{w,i} = \frac{(120,000 \text{ lb/hr})^2 (2 + 0.6(11))}{2 \left(4.17 \times 10^8 \frac{\text{ft lb}}{hr^2 \text{lb}_f} \left(\frac{201.9}{144} \text{ in.}^2 / \text{ft}^2\right) \frac{119}{144} \text{ in.}^2 / \text{ft}^2 \left(0.57 \frac{lb}{ft^3}\right)\right)} = 225 \text{ lb}_f / \text{ ft}^2 = 1.56 \text{ psi}$

d. $Re_t = 0.54$

e. $Re_b = 0.80$

f. $\Delta p_s = [(6 - 1)(238 \text{ lb/ft}^2)(0.80) + 6(225 \text{ lb/ft}^2)] \times 0.54 + 2(238 \text{ lb/ft}^2)(0.80) + (1 + 1/11)$

$= 2005 \text{ lb}_f / \text{ ft}^2 = 13.9 \text{ psi}$

This is the pressure drop for the case in which only vapor flows the entire length of the shell.

g. Correction for a condensing flow. From Fig. 3.25, we find that a totally condensing flow gives about 0.3 times as much pressure drop as that for vapor flowing the entire distance, so we estimate the actual shell side (nozzle-to-nozzle) pressure drop as 0.3 $(13.9 \text{ psi}) = 4.2 \text{ psi}$ which is slightly lower than the problem statement. However, it should be noted that these calculations are quite imprecise, and one could gain a small margin of safety (about 8 percent on the pressure drop) by specifying an exchanger with total tube length of 14 feet (with the same baffle spacing), giving an effective tube length of about 13.7 feet. Otherwise, the design seems to meet the requirements quite well.
# NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Heat transfer area. $A^*$, reference area; $A_{\text{fin}}$, fin heat transfer area; $A_{\text{eq}}$, effective area for a finned tube; $A_i$, $A_o$, inside and outside tube areas, respectively; $A_m$, mean wall heat transfer area; $A_{\text{root}}$, root heat transfer area for finned tube; $A_T$, total heat transfer area in exchanger.</td>
<td>ft²</td>
</tr>
<tr>
<td>a</td>
<td>Heat transfer area per foot of tube.</td>
<td>ft²/ft</td>
</tr>
<tr>
<td>a_{\text{fin}}</td>
<td>Area of one side of one fin.</td>
<td>ft²</td>
</tr>
<tr>
<td>$C_p, C_p$</td>
<td>Specific heat of vapor and coolant, respectively.</td>
<td>Btu/lb°F</td>
</tr>
<tr>
<td>$\overline{C}_{p,v}$</td>
<td>Mean specific heat of the vapor phase in a multi-component mixture.</td>
<td>Btu/lb°F</td>
</tr>
<tr>
<td>d</td>
<td>Diameter; $d_{\text{eq}}$, equivalent diameter of a finned tube in condensation; $d_i, d_o$, inside and outside diameters of a tube; $d_r$, root diameter of a finned tube.</td>
<td>in. or ft.</td>
</tr>
<tr>
<td>E</td>
<td>Mueller correction factor for total condensation, Eq. (3.50).</td>
<td>dimensionless</td>
</tr>
<tr>
<td>F</td>
<td>Correction factor for the logarithmic mean temperature difference (LMTD) to make it applicable to heat exchangers in which the flow is not entirely countercurrent or cocurrent.</td>
<td>dimensionless</td>
</tr>
<tr>
<td>$F, F'$</td>
<td>Fraction of tubes flooded in a condenser with integral subcooler.</td>
<td>dimensionless</td>
</tr>
<tr>
<td>$F_1, F_2$</td>
<td>Parameters in the Diehl-Koppany flooding velocity correlations, Eq. (3.30 and 3.31)</td>
<td>dimensionless</td>
</tr>
<tr>
<td>$F_3, F_4$</td>
<td>Correction factors for approximate sizing procedures.</td>
<td>dimensionless</td>
</tr>
<tr>
<td>$F_c$</td>
<td>Chen two-phase convective heat transfer multiplier.</td>
<td>dimensionless</td>
</tr>
<tr>
<td>$F_{vc}$</td>
<td>Parameter in Carpenter-Colburn equation for in-tube condensation, defined by Eq. (3.83).</td>
<td>lbm/ft hr²</td>
</tr>
<tr>
<td>f_i</td>
<td>Friction factor for pressure drop inside tubes.</td>
<td>dimensionless</td>
</tr>
<tr>
<td>g</td>
<td>Gravitational acceleration.</td>
<td>ft/sec²</td>
</tr>
<tr>
<td>$g_c$</td>
<td>Gravitational conversion constant.</td>
<td>32.2 lbm ft/lb sec²</td>
</tr>
<tr>
<td>G</td>
<td>Mass velocity (mass flow rate of fluid per unit cross-sectional area for flow). $G_v$ and $G_i$ are superficial mass velocities for vapor and liquid respectively; &quot;superficial&quot; means the values are calculated as if the given fluid were flowing alone, using the entire cross-sectional area. $G_{v,i}$, $G_{v,m}$, and $G_{v,o}$ are respectively the inlet, mean, and outlet superficial vapor mass velocities for</td>
<td>lbm/ft² hr</td>
</tr>
</tbody>
</table>
the Carpenter-Colburn correlation for intube condensation.

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>H</td>
<td>Fin height.</td>
<td>in. or ft.</td>
</tr>
<tr>
<td>h</td>
<td>Film heat transfer coefficient; $h_i$, $h_o$, inside and outside coefficients, respectively; $h_c$, average condensing coefficient; $h_{c,N}$, average coefficient for a vertical bank of $N$ tubes; $h_{sv}$, average sensible heat transfer coefficient for the vapor-gas mixture in multicomponent condensation.</td>
<td>Btu/hr ft$^2$°F</td>
</tr>
<tr>
<td>k</td>
<td>Thermal conductivity; $k_l$, liquid thermal conductivity; $k_v$, vapor thermal conductivity; $k_w$, thermal conductivity of tube wall.</td>
<td>Btu/hr ft$^2$°F</td>
</tr>
<tr>
<td>K</td>
<td>Parameter in Mueller total condensation analysis, Eq. (3.51)</td>
<td>dimensionless</td>
</tr>
<tr>
<td>L</td>
<td>Tube length.</td>
<td>ft.</td>
</tr>
<tr>
<td>$\bar{L}$</td>
<td>Equivalent length of fin.</td>
<td>ft.</td>
</tr>
<tr>
<td>LMTD</td>
<td>Logarithmic mean temperature difference.</td>
<td>°F</td>
</tr>
<tr>
<td>MTD</td>
<td>True mean temperature difference, $F$ (LMTD).</td>
<td>°F</td>
</tr>
<tr>
<td>m</td>
<td>Parameter in fin efficiency equation.</td>
<td>dimensionless</td>
</tr>
<tr>
<td>N</td>
<td>Number of tubes in a vertical row.</td>
<td>dimensionless</td>
</tr>
<tr>
<td>$N_f$</td>
<td>Number of fins per unit length.</td>
<td>fins/inch</td>
</tr>
<tr>
<td>$Nu_i$</td>
<td>Nusselt number for heat transfer inside a tube.</td>
<td>dimensionless</td>
</tr>
<tr>
<td>p</td>
<td>Pressure</td>
<td>lb/in$^2$ absolute</td>
</tr>
<tr>
<td>$P_{crit}$</td>
<td>Critical pressure of a fluid.</td>
<td>lb/in$^2$ absolute</td>
</tr>
<tr>
<td>$P_r$</td>
<td>Prandtl number for a fluid.</td>
<td>dimensionless</td>
</tr>
<tr>
<td>$P_R$</td>
<td>Reduced pressure, defined by Eq. (3.10).</td>
<td>dimensionless</td>
</tr>
<tr>
<td>$\Delta p$</td>
<td>Pressure drop; $\Delta p_{ent}$, pressure drop for entrance to a tube; $\Delta p$, pressure drop due to friction for flow inside a tube; $\Delta p$, pressure drop on the shell side, nozzle to nozzle.</td>
<td>lb/ft$^2$ or lb/in$^2$</td>
</tr>
<tr>
<td>$\Delta p_{TPF}$</td>
<td>Total pressure effect in a two-phase flow.</td>
<td>lb/in$^2$</td>
</tr>
<tr>
<td>$\Delta p_{m,TPF}$</td>
<td>Pressure effect due to momentum changes in two phase flow.</td>
<td>lb/in$^2$</td>
</tr>
<tr>
<td>$\left(\frac{dp}{dL}\right)_{f,c}$</td>
<td>Pressure gradient due to friction for liquid flowing in a conduit.</td>
<td>(lb/in$^2$)/ft</td>
</tr>
</tbody>
</table>
Pressure gradient due to friction for a two-phase mixture flowing in a conduit. \( \frac{dp}{dl} \) \(_{f,TPF} \) (lb/in\(^2\)/ft)

Pressure gradient due to friction for a vapor flowing in a conduit. \( \frac{dp}{dl} \) \(_{f,v} \) (lb/in\(^2\)/ft)

Pressure gradient due to hydrostatic effect in two-phase flow. \( \frac{dp}{dl} \) \(_{g,TPF} \) (lb/in\(^2\)/ft)

Pressure gradient due to momentum changes in two-phase flow. \( \frac{dp}{dl} \) \(_{m,TPF} \) (lb/in\(^2\)/ft)

Total pressure gradient in two-phase flow. \( \frac{dp}{dl} \) \(_{T,TPF} \) (lb/in\(^2\)/ft)

Q Heat load; \( Q_c \), condensing heat load; \( Q_{sc} \), subcooling heat load; \( Q_s \), heat heat load for sensible cooling of a vapor; \( Q_T \), total heat load in a condenser. Btu/hr

R\(_{fi}\), R\(_{fo}\) Fouling resistances, inside and outside surfaces, respectively. hr ft\(^2\)°F/Btu

R\(_{fin}\) Fin resistance to heat transfer. hr ft\(^2\)°F/Btu

R\(_{l}\), R\(_{v}\) Liquid and vapor volume fractions, respectively: the actual volume occupied by a given phase, divided by the total volume of the system. dimensionless

Re\(_{c}\) Condensate Reynolds number defined by Eq. (3.37). dimensionless

Re\(_i\) Reynolds number for flow inside a tube. dimensionless

s Spacing between fins. in. or ft.

T Temperature on the condensing side; \( T_s \), \( T_s' \), surface temperatures, actual and calculated, during desuperheating; \( T_{sat} \), saturation temperature of vapor; \( T_{sc} \), temperature of exiting subcooled condensate; \( T_w \), surface temperature in condensing zone. °F

t Temperature of coolant; \( t_1 \), \( t_2 \), inlet and outlet temperatures, respectively; \( t^* \), \( t_s^* \), mixed mean turnaround and exit temperatures in condensers with subcooling zones; \( t_{bo} \), \( t_{do} \), \( t_{so} \), \( t_{go} \), \( t_{bo} \), \( t_{do} \), \( t_{so} \), \( t_{go} \), \( t_{ao} \), \( t_{bo} \), \( t_{do} \), \( t_{so} \), \( t_{go} \), \( t_{fo} \), \( t_{go} \), \( t_{jo} \), \( t_{ko} \), local turnaround temperatures in condensers with subcooling zones. °F

U Overall heat transfer coefficient; \( U^* \), overall coefficient based on a reference area, \( A^* \); \( U_c \), \( U_{or} \), overall coefficients based upon the inside and outside tube areas, respectively; \( U_{sc} \), \( U_{or} \), and \( U_{sc} \), overall coefficients for the condensing zone, the sensible heat transfer zone, and the subcooling zone, respectively; \( U' \), and \( U_{or}' \), partial overall coefficients, including condensate film wall, fouling, and coolant resistances. Btu/hr ft\(^2\)°F

V\(_i\) In-tube fluid velocity. ft/sec or ft/hr
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$V'_c$</td>
<td>Superficial incipient flooding velocity of the vapor in a knockback condenser</td>
<td>ft/sec</td>
</tr>
<tr>
<td>$\dot{V}_l, \dot{V}_v$</td>
<td>Volume flow rates of liquid and vapor phases, respectively, in a two-phase flow.</td>
<td>ft$^3$/hr</td>
</tr>
<tr>
<td>$W$</td>
<td>Vapor mass flow rate. $W_{\text{in}}, W_{\text{out}}$, inlet and outlet vapor mass flow rates, respectively.</td>
<td>lb/hr</td>
</tr>
<tr>
<td>$W_t$</td>
<td>Condensate weight flow rate per tube defined in Eq. (3.46).</td>
<td>lb/hr</td>
</tr>
<tr>
<td>$w$</td>
<td>Coolant mass flow rate.</td>
<td>lb/hr</td>
</tr>
<tr>
<td>$x$</td>
<td>Distance from tube inlet.</td>
<td>ft</td>
</tr>
<tr>
<td>$X_o$</td>
<td>Outlet vapor quality.</td>
<td>dimensionless</td>
</tr>
<tr>
<td>$\Delta x_w$</td>
<td>Wall thickness.</td>
<td>in. or ft.</td>
</tr>
<tr>
<td>$Y$</td>
<td>Fin thickness.</td>
<td>in. or ft.</td>
</tr>
<tr>
<td>$Z$</td>
<td>Fraction of total heat duty that is vapor/gas sensible cooling.</td>
<td>dimensionless</td>
</tr>
</tbody>
</table>

**GREEK**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\lambda$</td>
<td>Latent heat of condensation.</td>
<td>Btu/lb</td>
</tr>
<tr>
<td>$\mu$</td>
<td>Viscosity. $\mu_l$, liquid phase viscosity; $\mu_s$, viscosity of fluid evaluated at surface temperature.</td>
<td>lb$_m$/ft hr</td>
</tr>
<tr>
<td>$\rho$</td>
<td>Density. $\rho_l$, liquid density; $\rho_v$, vapor density.</td>
<td>lb$_m$/ft$^3$</td>
</tr>
<tr>
<td>$\sigma$</td>
<td>Surface tension of a liquid.</td>
<td>dyne/cm</td>
</tr>
<tr>
<td>$\Gamma$</td>
<td>Condensate loading per foot of tube drainage perimeter, defined by Eq. (3.34).</td>
<td>lb$_m$/ft hr</td>
</tr>
<tr>
<td>$\delta$</td>
<td>Thickness of a condensate film.</td>
<td>in. or ft.</td>
</tr>
<tr>
<td>$\theta$</td>
<td>Angular orientation of a tube, Fig. 3.6.</td>
<td>degrees</td>
</tr>
<tr>
<td>$\Lambda$</td>
<td>Modified Baker parameter for two-phase flows in a horizontal tube.</td>
<td>lb./ft$^3$</td>
</tr>
<tr>
<td>$\Phi$</td>
<td>Fin efficiency.</td>
<td>dimensionless</td>
</tr>
</tbody>
</table>

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\[ \Phi_{gtt}^2 \quad \text{Multiplying factor for calculating total shell-side pressure drop in a condenser from vapor phase pressure drop.} \]

\[ \Phi_{tn}^2 \quad \text{Martinelli-Nelson two-phase flow friction pressure drop multiplier.} \]

\[ \chi_{tt} \quad \text{Martinelli-Nelson two-phase flow parameter defined by Eq. (3.15).} \]

\[ \Psi \quad \text{Modified Baker parameter for two-phase flows in a horizontal tube.} \]

\[ \Psi = \mu_t^{1/3} / \sigma_{t}^{2/3} \quad \text{ft}^{5/3} \text{cm} / \text{lb}_{m}^{1/3} \text{hr}^{1/3} \text{dyne} \]
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7. Nusselt, W., Zeits VDI, 60, 541, 569 (1916).


